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CYLINDER-HEAD TEMPERATURES AND COOLANT HEAT REJECTIONS

OF A MULTICYLINDER, LIQUID-COOLED ENGINE

OF 1710-CUBIC-INCH DISPLACEMENT

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SUMMARY

An extensive investigation of the cooling characteristics of a
multicylinder, liquid-cooled aircraft engine of 1710-cubic-inch dis-
placement was conducted at the NACA Cleveland laboratory. The
results of this investigation are presented showing the variation
of the cylinder-head temperature and the coolant heat rejection
with the pertinent engine and coolant variables. The data, which
were obtained on five engines, are presented for power outputs up
to 1860 brake horsepower, coolant flows from 50 to 320 gallons per
minute, and wide ranges of engine speed, manifold pressure, fuel-
air ratio, inlet-air temperature, ignition timing, exhaust pressure,
and coolant composition, temperature, and pressure. The ranges of
variables included operation into the boiling range of the coolant
for which various phenomena are described.

INTRODUCTION

The cooling characteristics of reciprocating aircraft engines
are an important factor for satisfactory performance at extreme
conditions of operation. A considerable amount of data on the
cooling characteristics of various air-cooled engines has been
published by various investigators but little data have been
published on the cooling characteristics of liquid-cooled engines.

An extensive investigation of the cooling characteristics of
liquid-cooled engines was therefore instituted at the NACA Cleve-
land laboratory in 1943. The initial phase of this investigation
consisted of a series of tests conducted on a single-cylinder
engine to provide data for a fundamental study of the heat-transfer
processes involved. These data, which isolate the effects of the
various engine and coolant variables on the cylinder-head tempera-
tures, are presented in reference 1. In reference 2, an analysis
based on the theory of nonboiling forced-convection heat transfer

was made of the cooling processes in a liquid-cooled engine and a
semiempirical method of correlating the cylinder-head temperatures
with the primary engine and coolant variables, similar to that pre-
sented in reference 3 for air-cooled engines, was derived and suc-
cessfully applied to the data of reference 1.

Following the investigations on the single-cylinder engine, a
comprehensive investigation of the cooling characteristics of a
multicylinder engine of 1710-cubic-inch displacement was conducted
during the period 1944-46. Both the cylinder-head temperatures and
the coolant heat rejection were determined for power outputs up to
1860 brake horsepower over wide ranges of engine speed, manifold
pressure, fuel-air ratio, inlet-air temperature, ignition timing,
exhaust pressure, and coolant flow, composition, temperature, and
pressure.

APPARATUS

Engines

The investigation was conducted on five V-1710 engines, which
shall be designated hereinafter engines A, B, C, D, and E. All
engines were standard production models of the same design with
regard to cooling and were unmodified except for the following:

1. Engine B was fitted with an aftercooler for part of the
investigation.

2. Engines C, D, and E were equipped with pistons machined
0.005 inch under standard diameter.

3. Engine D was equipped with a variable ignition-timing
device for part of the investigation.

These engines are liquid-cooled and have six cylinders in
each of two banks. The cylinder bore and stroke are 5.5 and
6.0 inches, respectively, and the displacement of the engines is
1710 cubic inches. The engine models used in the investigation
have a compression ratio of 6.65 and are fitted with a single-
stage gear-driven supercharger having an impeller diameter of
9.5 inches and a supercharger-to-engine-speed ratio of 9.6. The
standard ignition system is timed to fire the intake spark plugs
at 28° B.T.C. and the exhaust spark plugs at 34° B.T.C. The valve
overlap extends over a period of time equivalent to 74° travel of
the crankshaft. Each engine was equipped with an air-blast tube
to cool the exhaust spark plugs.

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General Engine Setup

A plan view of the engine setup is shown in figure 1.

Power measurement. - The engine was mounted on a dynamometer stand equipped with a 2000-horsepower air-gap eddy-current dynamometer. The engine speed was electronically controlled and was measured on a chronometric tachometer. A calibrated air-balanced diaphragm measured the torque transmitted to the dynamometer.

Combustion-air system. - Combustion air was supplied to the engine by the laboratory central system and was metered with an air orifice, which was installed in the inlet-air ducting according to A.S.M.E. specifications. The temperature of the air was controlled by passing it through either a heater or refrigeration unit and the air was cleaned by passing it through a filter. Thermocouples and pressure taps were installed at the orifice and at the carburetor inlet for measuring the temperature and the pressure of the air at these locations.

Exhaust system. - The engine exhaust gases were removed by means of the laboratory central exhaust system, which also provided the desired exhaust pressures. Water-jacketed exhaust stacks were used for all of the tests except those in which the exhaust pressure was varied. The stacks and collector used for the variable-exhaust-pressure runs were of the type used with a turbosupercharger installation on a typical fighter plane.

Coolant system. - A diagrammatic sketch of the coolant system is shown in figure 2(a). An auxiliary pump installed in the main coolant line and used in conjunction with bypass and throttle valves permitted control of the coolant flow independent of the engine speed. A venturi was used to measure the flow. A valve was located downstream of the venturi to raise the pressure in the venturi throat sufficiently to prevent cavitation during operation at high coolant flows and low coolant pressures. A compressed air and bleed-line combination connected to the coolant expansion tank was used to obtain desired engine coolant-outlet pressures. Vapor separators were installed in the engine coolant-outlet lines to remove air or any vapor that resulted from boiling of the coolant. In order to permit observation of the coolant condition, sight glasses were installed in the engine coolant-outlet vent lines and the vapor-separator vent lines.

The coolant temperature-control unit consisted of two aircraft-type coolers with a bypass line around them and a thermostatically

operated mixing valve installed at the junction of the main and
bypass lines. Water was used as the cooling medium and the flow
was measured with calibrated rotameters.

Oil system. - A diagrammatic sketch of the oil system is shown
in figure 2(b). The oil flow to the engine was measured by means
of an oil-weighing device incorporated in the oil-supply tank, as
described in reference 4. The oil temperature-control unit was
similar to that used in the coolant system.

Coolants, fuel, and oil. - The coolants used were AN-E-2 eth-
ylene glycol, 100-percent water, and mixtures of 30-, 50-, and
70-percent by volume of ethylene glycol in water. The specifica-
tion of AN-E-2 ethylene glycol on a weight basis is 94.5-percent
ethylene glycol, 2.5-percent triethanolamine phosphate, and
3.0-percent water. For convenience, AN-E-2 ethylene glycol will
be referred to as a "nominal" (by volume) mixture of 97-percent
ethylene glycol and 3-percent water. In order to inhibit cor-
rosion, 0.2 percent by volume of NaMBT (sodium mercaptobenzo-
thiazole) was added to all coolant mixtures.

For all runs, AN-F-28, Amendment-2, fuel was used and was
metered by calibrated rotameters. For knock-free engine operation
at high power, 3 percent by volume of xylidines was added and the
tetraethyl lead concentration was increased to 6 milliliters per
gallon.

The lubricating oil used throughout the tests was Navy 1120.

Instrumentation

Thermocouples for measuring engine temperatures. - The
cylinder-head thermocouple installation is shown in figures 3
and 4. Thermocouples were installed in the cylinder head between
the exhaust valves for all engines, in the cylinder head between
the intake valves for all of the engines except engine B, in the
cylinder head at the exhaust spark-plug boss for all of the engines
except engines B and E, and on the exhaust-valve guide, the exhaust
spark-plug gasket, and the intake spark-plug gasket for engine A
only. Several methods were employed to form the hot junctions of
the thermocouples located inside the cylinder head. The most
satisfactory method was to silver-solder the two leads of the
thermocouple to a small brass plug; the plug was then peened into
a number 56 drilled hole $\frac{1}{16}$ inch deep located at the bottom of a
$\frac{1}{8}$-inch drilled hole. The leads were packed in place in the $\frac{1}{8}$-inch

hole with porcelain cement and were covered on the outside of the
engine with $\frac{1}{8}$-inch thin-wall stainless-steel tubing to prevent
breakage. The thermocouple holes were located with drill jigs in
each cylinder head and in each engine.

Cylinder-barrel thermocouples were installed on engine A, as
shown in figure 5. Thermocouples were located at the top of each
cylinder barrel on the intake and exhaust sides, and at the middle
of each barrel on the exhaust side. In addition, cylinders 1 and 6
of both banks had thermocouples located at the bottom of the barrel
on both the intake and exhaust sides. For all locations, the hot
junction of each thermocouple was made by spot-welding the wires to
the barrel. The hot junction was then covered with an insulating
varnish, which was baked dry. The leads were insulated from the
barrel by means of a flexible glass sleeve and securely staked at
intervals by short pieces of Nichrome wire spot-welded to the
barrel. The leads were gathered together and brought out through
pressure-tight fittings screwed into the cylinder on the intake
side, as shown in figure 4.

All thermocouples used for cylinder-temperature measurements
were made of 24-gage iron-constantan wire and were connected to
self-balancing direct-reading potentiometers.

Thermocouples for measuring liquid temperatures. - The tempera-
tures of the coolant, the oil, and the cooling water were measured
at the locations shown in figure 2 by both copper-constantan and
iron-constantan thermocouples. The copper-constantan thermocouples
were connected to a portable precision-type potentiometer equipped
with a sensitive light-beam galvanometer and gave an accurate
measurement of the temperature differences across the engine and
the coolers. The iron-constantan thermocouples were connected to
a self-balancing direct-reading potentiometer and gave an immediate
indication of the temperature.

Pressure taps. - Pressure taps were installed in the coolant
and oil systems at the locations shown in figure 2. In order to
determine the coolant-flow distribution through the engine, pres-
sure taps were also installed across the standard coolant-metering
orifices in the cylinder jacket (figs. 6 and 7). These coolant-
metering orifices were calibrated in a bench setup.

PROCEDURE AND METHODS

Conditions

Cylinder temperatures and coolant heat rejections were deter-
mined over the following range of conditions:

Engine speed, rpm . . . . 1200 - 3200
Manifold pressure, in. Hg absolute . . . . 21.0 - 73.0
Engine power, bhp . . . . 275 - 1860
Charge flow (air plus fuel), lb/sec . . . . 0.73 - 4.40
Fuel-air ratio . . . . 0.062 - 0.118
Carburetor-air temperature, °F . . . . -43 - 184
Ignition timing, deg B.T.C.
  Intake spark plugs . . . . 8 - 48
    Exhaust spark plugs . . . . 14 - 54
Exhaust pressure, in. Hg absolute . . . . 6.0 - 61.0
Coolant flow, gal/min . . . . 50 - 320
Average coolant temperature, °F . . . . 149 - 302
Engine coolant-outlet pressure, lb/sq in. gage . . . . . . 10 - 45

Data were obtained for coolants composed of 100-percent water and
mixtures of 30, 50, 70, and 97 percent by volume of ethylene glycol
in water.

In order to isolate the effect of the engine and coolant
variables on the cylinder temperatures and coolant heat rejection,
one of these parameters was varied in each test while, in general,
all the others were maintained constant. A complete summary of the
conditions of the investigation is presented in table I, and all of
the pertinent conditions for each test except ignition timing,
exhaust pressure, and oil-inlet temperature are also given in the
legend of each figure. These three items are omitted from the
figures because they were constant (see table I for values) for
all of the tests, except those in which they were the primary
variable.

The charge flow (air plus fuel) rather than the brake horse-
power was held constant for most of the tests in which one of the
other parameters was varied. For the variable engine-speed and
variable exhaust-pressure tests, the charge flow was maintained
constant by varying the manifold pressure accordingly. For the
variable charge-flow runs, two methods were employed to vary the
charge flow: (1) the engine speed was held constant while the
manifold pressure was varied, and (2) the manifold pressure was
held constant while the engine speed was varied.

The coolant flow was varied at several engine powers for several combinations of coolant temperature, pressure, and composition. The highest coolant flow attainable in each case was limited either by the cavitation characteristics of the pumps at low coolant pressure or the capacities of the pumps at high coolant pressure. The lowest coolant flow attainable was limited by severe boiling of the coolant in the engine. Depending upon the type of tests conducted, either the average or the outlet coolant temperature was held constant.

The ethylene-glycol concentration of the coolant was determined from the boiling point and the specific gravity of samples taken at intervals throughout the investigation.

Calculations

Temperature averages. - The average cylinder temperature for each thermocouple location was determined by averaging the temperatures measured at that location in all 12 cylinders. The average coolant temperature was taken as the arithmetic mean of the measured inlet and outlet temperatures in each case.

Heat rejected to the coolant. - The heat rejected to the coolant was determined by two methods: (1) from the measured temperature rise and flow of the coolant through the engine and (2) from the measured temperature rise and flow of the cooling water. The heat rejected to the coolant is presented on the basis of method 2 because at the low coolant flows, when large amounts of vapor were formed, method 1 would not include the heat of vaporization and at the high coolant flows difficulty was experienced in accurately measuring the small temperature rise incurred in the passing of the coolant through the engine. The external heat losses from the coolant piping were estimated to be about 2 percent of the heat rejected and the heat rejections calculated on the basis of method 2 are corrected for these losses.

RESULTS AND DISCUSSION

Relations between Cylinder Temperatures

The relation between the average temperature of the 12 cylinders in the cylinder head between the exhaust valves and the temperature of the hottest cylinder (maximum temperature) measured for the same location is shown in figure 8. All the data obtained on the five engines for the various operating conditions

investigated are presented, and the scatter is within ±10° F. A
linear variation is noted for the entire range of temperatures
measured with the maximum temperature ranging from 10° to 20° F
higher than the average temperature.

The variation of the average temperatures for the various
locations in the cylinders with the average cylinder-head tempera-
ture between the exhaust valves is shown in figures 9 to 16 for
all the data at the various operating conditions. A linear rela-
tion exists in each figure and the mean scatter of the data is
about ±15° F. The average temperature at the various locations in
the cylinder are lower than the temperature between the exhaust
valves by the following amounts:

Location	Temperature difference (°F)	Figure
Between intake valves	60 to 110	9
Exhaust spark-plug boss	50 to 75	10
Exhaust spark-plug gasket	140 to 155	11
Intake spark-plug gasket	145 to 180	12
Exhaust valve guide	145 to 170	13
Middle of barrel, exhaust side	170 to 210	14
Top of barrel, intake side	150 to 170	15
Top of barrel, exhaust side	160 to 190	16

Because of the linear relation existing between the temperatures
in the various locations in the cylinders, the variation of only
one of them with the pertinent engine and coolant variables is
presented. The average cylinder-head temperature between the
exhaust valves was chosen for this purpose because it was meas-
ured in the hottest region of the cylinder head and thus is most
indicative of critical cooling conditions.

Effect of Engine Variables on Average Cylinder-Head

Temperature between Exhaust Valves

Charge flow and engine power. - The effect of charge flow on
the average cylinder-head temperature between the exhaust valves
is presented in figure 17 for two fuel-air ratios, a range of
carburetor-air temperatures, and for both variable-speed and

variable-manifold-pressure data. Although a straight line may be
drawn through all the data for each fuel-air ratio, a close inspec-
tion shows that the cylinder-head temperature for the variable-
speed data increases slightly faster than for the variable-manifold-
pressure data for the same change in charge flow. The more rapid
increase of the cylinder-head temperature with engine speed is a
result of an attendant increase in the inlet-manifold temperature
caused by an increase in the temperature rise across the super-
charger. The straight lines, which represent all the data, show
an increase in average cylinder-head temperature with charge flow
of about 50° F per pound per second increase in charge flow.

In order to permit a determination of the variation of the
cylinder-head-temperature data of figure 17 with engine power, the
variation of engine power with charge flow is presented in figure 18
for engine speeds of 2600, 3000, and 3200 rpm. For each engine
speed presented, the engine power increased linearly with charge
flow and an increase in engine power of 100 brake horsepower
resulted in approximately a 10° F increase in head temperature.

Carburetor-air temperature. - The effect of carburetor-air
temperature on the average cylinder-head temperature between the
exhaust valves is presented in figure 19 for four values of charge
flow. For all charge flows presented, the increase in the average
cylinder-head temperature with carburetor-air temperature was
linear and amounted to about 3° F for an increase of 100° F in
carburetor-air temperature.

Engine speed. - The variation of the average cylinder-head
temperature between the exhaust valves with engine speed is pre-
sented in figure 20 for several combinations of charge flow and
carburetor-air temperature. For these tests, the charge flow was
maintained constant by varying the manifold pressure as the engine
speed was varied.

The increase in the average cylinder-head temperature with
engine speed was linear for each combination of charge flow and
carburetor-air temperature, and the maximum increase amounted to
about 8° F for an increase in engine speed of 1000 rpm. As was
previously indicated, this variation of average cylinder-head
temperature with engine speed is attributed to the increase in
manifold temperature caused by an increase in the temperature
rise across the supercharger with engine speed.

Fuel-air ratio. - The effect of fuel-air ratio on the average
cylinder-head temperature between the exhaust valves is presented
in figure 21 for several engine operating conditions. The shapes

of the curves are similar, with a maximum cylinder-head temperature
occurring at a fuel-air ratio of 0.067. This value of fuel-air
ratio is approximately equal to that for the stoichiometric mixture.
For an increase in fuel-air ratio from 0.067 to 0.118, the average
cylinder-head temperature decreased approximately 65° F for each
engine operating condition.

Ignition timing. - The variation of the average cylinder-head
temperature between the exhaust valves with ignition timing for
two values of engine speed is shown in figure 22. A minimum point
in the variation occurred for an ignition setting corresponding to
an exhaust spark-plug timing of about 30° B.T.C. at an engine
speed of 2600 rpm and about 34° B.T.C. at an engine speed of
3000 rpm. The maximum rate of increase in the average cylinder-
head temperature with advanced spark timing occurred at an engine
speed of 3000 rpm and amounted to about 20° F for an advance in
the exhaust spark-plug timing from the minimum point (34° B.T.C.)
to 52° B.T.C.

Exhaust pressure. - The variation of the average cylinder-head
temperature with exhaust pressure for the condition where the
charge flow was held constant by varying the manifold pressure is
presented in figure 23(a) for several fuel-air ratio and power
conditions. For each condition, the increase in cylinder-head
temperature with exhaust pressure is linear and is slightly
greater at the lowest fuel-air ratio. For an increase in exhaust
pressure of 40 inches of mercury, the increase in the average
cylinder-head temperature ranged from about 23° F for a fuel-air
ratio of 0.100 to about 32° F for a fuel-air ratio of 0.063.

The results obtained when the manifold pressure was main-
tained constant with attendant variation in charge flow are pre-
sented in figure 23(b). For this condition the change in the
average cylinder-head temperature between the exhaust valves with
exhaust pressure was negligible over the entire range of exhaust
pressures investigated. This negligible change is due to the
fact that the previously determined effect of the increased
exhaust pressure was offset by the effect of the corresponding
decrease in charge flow.

Variation of cylinder-head temperatures with engine running
time. - The variation of the average cylinder-head temperature
with engine running time for several thermocouple locations is
shown in figure 24. The data presented were obtained at a refer-
ence operating condition from time to time during the course of
the investigation on engine D. The coolant used for this refer-
ence operating condition was composed of 30-percent ethylene

glycol and 70-percent water. Coolants of various compositions were
used, however, between the runs at the reference operating condi-
tion. For an increase in engine running time from approximately
20 to 150 hours, the average cylinder-head temperature between the
exhaust valves increased about 20° F, at the exhaust spark-plug
boss increased about 6° F, and between the intake valves showed no
change. A leveling off of the cylinder-head temperatures in the
exhaust side of the cylinder head after about 100 hours running
time is noted. A close inspection of the coolant passages of a
scrapped cylinder block revealed scale deposits on the exhaust side
of the cylinder head but none on the intake side. The increase of
the temperatures in the exhaust side of the cylinder head is attri-
buted to these scale deposits. The greater increase with engine
running time of the cylinder-head temperature between the exhaust
valves as compared with that at the spark-plug boss is probably the
result of a greater amount of scale formation in the hotter region
of the cylinder head. The leveling off of the exhaust-side tem-
peratures after about 110 hours running time is probably the result
of a stabilization in the amount of scale formed in this region of
the cylinder.

The data presented in all the other figures are not corrected
for the effect of engine running time so that temperatures obtained
for exactly the same operating conditions but at different running
times may be slightly different. The temperature variations pre-
sented in each figure were not greatly influenced by engine running
time, because all runs of a series were run off in a relatively
short interval of engine running time.

Coolant-Flow Distribution

The coolant-flow distribution in the cylinder banks is pre-
sented in figure 25 on the basis of the percentage of the total
coolant flow to the engine. The percentage flow distribution was
the same for flows of 144 gallons per minute and 255 gallons per
minute. As a result, changes in the total coolant flow to the
engine will cause proportional changes in the coolant flow over
each cylinder head and barrel. Only the variations in the total
coolant flow to the engine will therefore be considered in the
following discussions of the effects of coolant flow on cooling.

Effect of Coolant Variables on Average Cylinder-Head Temperatures

between Exhaust Valves

The effects of the coolant variables investigated on the
average cylinder-head temperature between the exhaust valves are
shown in figures 26 to 31. Each variable rather than each figure
will be discussed individually, because the effects of certain
variables are illustrated in more than one figure.

Coolant composition. - A comparison of the average cylinder-
head temperature between the exhaust valves for coolant mixtures
varying in composition from 100-percent water to 97-percent ethyl-
ene glycol and 3-percent water, is presented in figure 26 for two
engine power conditions and a range of coolant flows. These data
were obtained for constant average coolant temperature and non-
boiling coolant conditions. The boiling or nonboiling condition
of the coolant was determined by observation of the coolant in the
sight glasses. The change in the average cylinder-head tempera-
ture obtained by changing the coolant composition was the same at
both engine-power conditions and was essentially independent of
coolant flow for all changes in coolant composition except when
the coolant composition was changed to or from 97-percent ethylene
glycol and 3-percent water. In this case the change in head tem-
perature is greater at the low flows. For all conditions, an
increase in the ethylene-glycol concentration of the coolant
resulted in an increase in the average cylinder-head temperature.
A comparison of the average cylinder-head temperatures obtained
when using 100-percent water as the coolant with those obtained
when using 97-percent ethylene glycol and 3-percent water shows
differences of about 60° F at a coolant flow of 300 gallons per
minute, and about 90° F at a coolant flow of 50 gallons per minute.

The average cylinder-head temperatures between the exhaust
valves for several coolant mixtures are shown in figures 27 and 28
for ranges of coolant flow at engine powers of 1000 and 1800 brake
horsepower, respectively. These data were obtained for constant
coolant-outlet temperatures, and are presented for nonboiling and
boiling coolant conditions. The relative effect of coolant com-
position obtained at these two power conditions are similar to
and of the same order of magnitude as those noted for figure 26
even though the coolant boiled under certain conditions of coolant
flow, temperature, and pressure.

Coolant flow. - The variation of the average cylinder-head
temperature between the exhaust valves with coolant flow for
constant average coolant temperature and nonboiling coolant

conditions is illustrated in figure 26. For all coolant mixtures, the average cylinder-head temperature increased as the coolant flow was decreased. A decrease in coolant flow from 300 to 50 gallons per minute resulted in a maximum increase in the average cylinder-head temperature amounting to approximately 70° F when using the coolant containing 97-percent ethylene glycol and an increase of approximately 38° F when using any one of the other coolants investigated. For these conditions the effect of coolant flow is nearly independent of engine power.

Inspection of figure 27 indicates that for constant coolant-outlet temperature and nonboiling coolant conditions, the increase in average cylinder-head temperature with decreasing coolant flow is similar but slightly less than that shown in figure 26 for constant average coolant temperature. This smaller increase in cylinder-head temperature is largely the result of the decrease in average coolant temperature accompanying the decreased coolant flow. This decrease in average coolant temperature when the coolant-outlet temperature is held constant amounted to about 6° F for a decrease in coolant flow from 250 to 100 gallons per minute and resulted from the constancy of the coolant heat rejection with changes in coolant flow.

The data presented in figure 28 for two coolant-outlet temperatures and for both nonboiling and boiling coolant conditions indicate that the rate of increase of the cylinder-head temperature with coolant flow is appreciably affected by the coolant temperature and pressure. A decrease in coolant flow from 300 to 60 gallons per minute resulted in increases in the average cylinder-head temperature from about 8° to 30° F depending upon the coolant operating conditions.

During these tests, three different modes of heat transfer were encountered and the existence of a particular mode was dependent upon the combination of coolant temperature, flow, and pressure. The first mode was characterized by the absence of boiling of the coolant and by a relatively large increase in the average cylinder-head temperature with a reduction in coolant flow. This mode is exemplified by the high-pressure and the high coolant-flow data presented in figure 28 for the coolant mixture of 30-percent ethylene glycol and 70-percent water at a constant coolant-outlet temperature of 250° F and also by all of the data presented in figures 26 and 27.

The second mode was characterized by the presence of moderate amounts of boiling as indicated by the passage of vapor through the vent-line sight glasses and by a relatively small increase in

the average cylinder-head temperatures between the exhaust valves
with reduction in coolant flow. These phenomena are illustrated in
figure 28 by all the data for both coolants at a coolant-outlet
temperature of 275° F and by the data for the coolant mixture of
30-percent ethylene glycol and 70-percent water at a coolant-outlet
pressure of 25 pounds per square inch gage and a coolant-outlet
temperature of 250° F for coolant flows less than 120 gallons per
minute. In these ranges of operation, boiling of the coolant
evidently suppressed the normal tendency of the cylinder-head
temperatures to increase as the coolant flow was decreased. The
decrease in the average coolant temperature of about 15° F with
decreased coolant flow from about 120 to 50 gallons per minute also
contributed to the suppression of the normal tendency of the
cylinder-head temperature to increase as the flow was decreased.

The third mode was marked by the presence of large amounts of
boiling and by a rapid increase in cylinder-head temperatures with
a reduction in coolant flow. This effect is shown by the curves
in figure 28 for the coolant mixture of 30-percent ethylene glycol
and 70-percent water at a coolant-outlet temperature of 250° F and
a coolant-outlet pressure of 15 pounds per square inch gage. The
difference between the second and third modes is similar to the
difference between nuclear and film-type boiling phenomena wherein
transition from nuclear to film-type boiling results in decreased
heat transfer because of the insulating qualities of the vapor
film formed. Besides being dependent upon the combination of
coolant-outlet pressure and temperature conditions, the range of
flow wherein each of the foregoing effects predominate would also
be expected to be dependent upon coolant composition and engine
power level.

The different modes of heat transfer are further illustrated
in figure 29 wherein the effects of coolant flow on the cylinder-
head temperature between the exhaust valves is illustrated for
constant average coolant temperature and for both nonboiling and
boiling coolant conditions. Data are presented for a coolant
mixture of 30-percent ethylene glycol and 70-percent water and
for two combinations of coolant temperature and coolant-outlet
pressure. The variation of both the average and the maximum
(hottest) cylinder-head temperature are shown and the results are
similar in each case. For both average coolant temperatures pre-
sented, the increase in the cylinder-head temperature for a
decrease in coolant flow from about 220 to 120 gallons per minute
was of the same order of magnitude as that shown in figures 26
and 27 for nonboiling coolant conditions. Further reduction of
the coolant flow from 120 to 110 gallons per minute resulted in a

slight decrease in cylinder-head temperature, which is attributed
to a sudden transition in the mechanism of heat transfer from
mode 1 to mode 2. (This transition was more gradual and did not
result in a decrease in cylinder-head temperature for the case
where the coolant-outlet temperature was held constant (fig. 28),
probably because of the attendant decrease in average coolant tem-
perature with coolant flow.) Further decrease in the coolant flow
resulted in a more rapid rise in cylinder-head temperature than was
obtained at the high coolant flows. This rapid rise was probably
the result of another transition in the mechanism of heat transfer
from mode 2 to mode 3.

To recapitulate, the data presented in figures 26 to 29 indi-
cate that, in general, for any large decrease in coolant flow (at
least 100 gal/min) the average cylinder-head temperature between
the exhaust valves increased the greatest amount when the average
coolant temperature was held constant and there was no apparent
boiling of the coolant; it increased the smallest amount when the
coolant-outlet temperature was held constant and there was moderate
boiling of the coolant. The greatest rate of increase in cylinder-
head temperature with decrease in coolant flow occurred at low
coolant flows and low outlet pressures when boiling of the coolant
was severe and large amounts of insulating vapor were formed.

Average coolant temperature. - The effect of average coolant
temperature on the average cylinder-head temperature between the
exhaust valves for coolant mixtures of 30- and 97-percent ethylene
glycol in water is shown in figure 30. For both of these coolants
and for the range of engine powers presented, the increase of the
average cylinder-head temperature with average coolant temperature
was linear and amounted to approximately 85° F for an increase in
average coolant temperature of 100° F.

Coolant-outlet pressure. - The variation of the cylinder-head
temperature between the exhaust valves with coolant-outlet pressure
is shown in figure 31 for an engine power of 1800 brake horsepower,
a coolant mixture of 30-percent ethylene glycol and 70-percent
water, and an average coolant temperature of 226° F. Both the
average and maximum cylinder-head temperatures are presented at
coolant flows of 60 and 200 gallons per minute. At the high cool-
ant flow, the rate of reduction of the cylinder-head temperatures
with decreased coolant pressure was uniform over the entire range
of pressures covered (fig. 31(b)), which indicates a gradual
increase in the boiling of the coolant. At the low coolant flow,
the rate of reduction of the cylinder-head temperature with
coolant-outlet pressure increased as the outlet pressure was
decreased from 45 to 25 pounds per square inch gage (fig. 31(a)).

Further reduction of the coolant pressure resulted in a leveling off and then an increase in the maximum cylinder-head temperature. This leveling off and increase is attributed to a transition from nuclear to film-type boiling such as occurred under certain conditions when the coolant flow was reduced. For the entire range of coolant pressures, the cylinder-head temperatures varied less than 21° F. Effects of coolant-outlet pressure on the cylinder-head temperature, similar to those just discussed, are also shown in figure 28.

During operation in the boiling range, severe boiling (film type) with a consequent rise in cylinder-head temperature may be encountered when reducing either the coolant flow or the coolant pressure. Although cylinder-head temperatures that are considered excessive were not obtained in the present investigation, it is likely that further reduction of either coolant flow or pressure below the lowest values investigated would result in cylinder temperatures that are considered unsafe.

Effect of Engine Variables on Heat Rejection to Coolant

Heat balance. - The heat balance between the coolant and the coolant cooling water is presented in figure 32. Fair agreement is seen to exist between the results of the two methods of calculating the heat rejection. In the region of high coolant heat rejections, most of the data fall above the match line, which indicates that, in general, a lower value of heat rejection was obtained when calculated on the coolant basis. This lower value of heat rejection is largely due to the fact that on the coolant basis the heat of vaporization would not be included for runs during which the coolant boiled.

Charge flow and engine power. - The variation of the coolant heat rejection with charge flow for two values of fuel-air ratio obtained when the engine speed was held constant and the manifold pressure was varied is shown in figure 33(a). For both fuel-air ratios, the heat rejection to the coolant increased linearly with charge flow; the increase amounted to approximately 72 Btu per second for an increase of 1 pound per second in charge flow.

The variation of the heat rejection to the coolant with charge flow when the manifold pressure was held constant and the engine speed was varied is presented in figure 33(b). The heat rejection increased more rapidly with charge flow for the variable-engine-speed data than for the variable-manifold-pressure data illustrated in figure 33(a). This more rapid increase is attributed largely

to the increase in inlet-manifold temperature with engine speed
(discussed in connection with the effect of charge flow and also
engine speed on the average cylinder-head temperature) and may also
be due in part to an increase in the cylinder-wall friction with
engine speed.

Carburetor-air temperature. - The variation of the coolant
heat rejection with carburetor-air temperature is presented in
figure 34 for two values of charge flow. Straight-line curves,
similar to those for the corresponding cylinder-head temperature
variation, were found to fit the data. The increase in the coolant
heat rejection for an increase of 100° F in carburetor-air tem-
perature amounted to about 15 Btu per second for both values of
charge flow.

Engine speed. - The variation of the heat rejection to the
coolant with engine speed is presented in figure 35 for several
engine operating conditions and charge flows. For the engine
operating conditions presented, the coolant heat rejection
increased linearly with engine speed, approximately 25 Btu per
second for an increase in engine speed of 1000 rpm. As previously
mentioned, this effect of engine speed on the coolant heat rejec-
tion is largely the result of an increase in inlet manifold tem-
perature with engine speed and may also be due in part to an
increase in the cylinder-wall friction with engine speed.

Fuel-air ratio. - The effect of fuel-air ratio on the heat
rejection to the coolant is presented in figure 36 for two engine
power conditions. As for the corresponding cylinder-head tem-
perature variation (fig. 21), the shapes of the curves for the
two power conditions are similar with a maximum heat rejection
occurring at a fuel-air ratio of 0.067. For an increase in fuel-
air ratio from 0.067 to 0.118, the heat rejection to the coolant
decreased approximately 100 Btu per second.

Ignition timing. - The relation between the heat rejection
to the coolant and ignition timing for two values of engine speed
is shown in figure 37. A minimum point, similar to that for the
cylinder-head temperature (fig. 22) occurred at an exhaust spark-
plug timing of about 30° B.T.C. for an engine speed of 2600 rpm
and about 34° B.T.C. for an engine speed of 3000 rpm. The maximum
change in the coolant heat rejection occurred at an engine speed
of 3000 rpm and amounted to an increase of about 35 Btu per second
for an advance in the exhaust spark-plug timing from the minimum
point (34° B.T.C.) to 52° B.T.C.

Exhaust pressure. - The effect of exhaust pressure on the heat rejection to the coolant is shown in figure 38. The curves are similar to those of figure 23 wherein the variation of the average cylinder-head temperature with exhaust pressure is presented. The relation between the coolant heat rejection and the exhaust pressure when the charge flow was held constant by varying the manifold pressure is presented in figure 38(a) for several fuel-air ratio and power conditions. For each condition, the increase in the heat rejection with exhaust pressure is linear and is slightly greater at the lowest fuel-air ratio. For an increase in exhaust pressure of 40 inches of mercury, the increase in coolant heat rejection ranged from 37 Btu per second for a fuel-air ratio of 0.100 to 58 Btu per second for a fuel-air ratio of 0.063.

The variation of the coolant heat rejection shown in figure 38(b) for constant manifold pressure and variable charge flow was negligible over the entire range of exhaust pressures investigated. This effect is due to the fact that the previously determined effect of the increased exhaust pressure was offset by the effect of the accompanying decreased charge flow.

Variation of the heat rejection with engine running time. - No measurable change occurred in the heat rejection to the coolant with engine running time.

Effect of Coolant Variables on Coolant Heat Rejection

The variation of the coolant heat rejection with the different coolant variables investigated is presented in figures 39 to 41. As for the plots of the cylinder-head temperatures, the effects of some variables are illustrated in more than one figure and therefore each variable rather than each figure will be individually discussed.

Coolant composition. - The effect of coolant flow on the heat rejected to the coolant for coolant mixtures varying in composition from 100-percent water to 97-percent ethylene glycol and 3-percent water is shown in figure 39. These data were obtained for constant average coolant temperatures and essentially nonboiling coolant conditions. For all conditions investigated, an increase in the ethylene-glycol concentration of the coolant resulted in a decrease in the coolant heat rejection; this decrease was essentially independent of coolant flow. For a change in coolant composition from 100-percent water to 97-percent ethylene glycol and 3-percent water, the decrease in coolant heat rejection was about 22 Btu per second.

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The heat rejection to the coolant at an engine power of
1800 brake horsepower when using coolant mixtures of 30- and
50-percent ethylene glycol in water is presented in figure 40 for
a range of coolant flows at two coolant-outlet temperatures. For
these conditions, during which there was considerable boiling of
the coolant for most of the runs, the difference between the
coolant heat rejections for the two coolants is approximately the
same as for the case where there was no boiling of the coolant
(fig. 39). The scatter of the data, however, is greater than
that for figure 39, probably because of difficulties in accurately
measuring the heat rejection under boiling conditions.

Coolant flow. - The variation of the coolant heat rejection
with coolant flow shown in figure 39, for constant average coolant
temperature and nonboiling coolant conditions, was essentially
independent of coolant composition. For a reduction in coolant
flow from 300 to 50 gallons per minute, the coolant heat rejection
decreased approximately 25 Btu per second.

At constant coolant-outlet temperature and for both boiling
and nonboiling coolant conditions (fig. 40) the decrease in cool-
ant heat rejection with decreased coolant flow was less than half
that shown in figure 39 for the entire range of coolant flows
investigated. This smaller decrease is probably the result of
both an increase in the intensity of boiling and a decrease in
the average coolant temperature, which accompanied the decrease
in coolant flow for these conditions. Thus, as for the cylinder-
head temperatures, the effect of coolant flow on the coolant heat
rejection is dependent upon boiling or nonboiling coolant condi-
tions and whether the average or outlet coolant temperature is
held constant.

Coolant temperature. - The variation of the coolant heat
rejection with average coolant temperature for coolant mixtures
of 30- and 97-percent ethylene glycol in water for a range of
powers from 785 to 1450 brake horsepower is shown in figure 41.
For the range of conditions investigated, an increase in the
average coolant temperature resulted in a linear decrease in the
coolant heat rejection of approximately 50 Btu per second per
100° F increase in coolant temperature.

Coolant pressure. - The coolant heat rejection data plotted
against coolant flow in figure 40 were obtained for several
coolant-outlet pressures and are keyed accordingly. No signifi-
cant trends or effects of the coolant-outlet pressure on the
coolant heat rejection are evident for the range of conditions
investigated.

SUMMARY OF RESULTS

From an investigation of the cooling characteristics of a multicylinder, liquid-cooled aircraft engine of 1710-cubic-inch displacement over wide ranges of engine and coolant conditions, the following results were obtained:

1. Both the cylinder-head temperature between the exhaust valves and the coolant heat rejection increased with the following variables:

(a) Linearly with charge flow (air plus fuel); the increase amounted to about 50° F and 72 Btu per second, respectively, for an increase in charge flow of 1 pound per second.

(b) Linearly with carburetor-air temperature; the increase amounted to about 3° F and 15 Btu per second, respectively, for 100° F increase in air temperature.

(c) Linearly with engine speed when the charge flow was held constant (a result of an attendant increase in the manifold temperature resulting from an increase in the temperature rise across the supercharger with engine speed); the increase amounted to about 8° F and 25 Btu per second, respectively, for an increase in engine speed of 1000 rpm.

(d) As the fuel-air mixture was leaned to a fuel-air ratio of about 0.067 and then decreased with further leaning. A decrease in fuel-air ratio from 0.118 to 0.067 resulted in increases of 65° F and 100 Btu per second, respectively.

(e) As the ignition timing was either increased or decreased from a particular value for each engine speed. The maximum rate of increase occurred at an engine speed of 3000 rpm and amounted to about 20° F and 35 Btu per second, respectively, for an advance in the exhaust spark-plug timing from the minimum point (34° B.T.C.) to 52° B.T.C.

(f) Linearly with exhaust pressure when the charge flow was held constant (variable manifold pressure); for an increase in exhaust pressure of 40 inches of mercury the increase ranged from 23° F and 37 Btu per second, respectively, at a fuel-air ratio of 0.100 to 32° F and 58 Btu per second, respectively, at a fuel-air ratio of 0.063. There was no significant change in either head temperature or coolant heat rejection when the manifold pressure was held constant (variable charge flow).

2. The cylinder-head temperature between the exhaust valves increased and the coolant heat rejection decreased:

(a) With an increase in the ethylene glycol concentration of the coolant. For a change in coolant composition from 100-percent water to 97-percent ethylene glycol and 3-percent water, the cylinder-head temperature increased about 60° F at a coolant flow of 300 gallons per minute and about 90° F at 50 gallons per minute, and the coolant heat rejection decreased about 22 Btu per second for all coolant flows.

(b) With a decrease in coolant flow; the magnitude of the change depended upon whether or not the coolant boiled and whether the average or outlet coolant temperature was held constant. For a decrease in coolant flow from 300 to 50 gallons per minute a maximum increase in the cylinder-head temperature of about 70° F and a maximum decrease in the coolant heat rejection of about 25 Btu per second occurred for constant average coolant temperature and non-boiling coolant conditions when using a coolant composed of 97-percent ethylene glycol and 3-percent water.

(c) Linearly with an increase in average coolant temperature; the change amounted to about 85° F and 50 Btu per second, respectively, for an increase of 100° F in coolant temperature.

3. A linear relation existed among the average temperatures for each location in all of the 12 cylinders and also between the maximum and average temperature for each location.

4. The variation of the cylinder-head temperature with coolant-outlet pressure depended upon the mode of heat transfer. For the entire range of pressures tested, the maximum change in head temperature with coolant pressure was less than 21° F.

5. The temperature in the cylinder head between the exhaust valves increased about 20° F for an increase in engine operating time from 20 to 150 hours. This increase is attributed to a progressive scale buildup on the coolant passages in the exhaust side of the cylinder head.

6. No significant variation of the coolant heat rejection with either coolant pressure or engine running time could be detected.

Flight Propulsion Research Laboratory,
    National Advisory Committee for Aeronautics,
        Cleveland, Ohio, December 29, 1947.

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TABLE I - SUMMARY

Variable or type of test	Engine power (bhp)	Engine speed (rpm)	Manifold pressure (in. Hg abs.)	Charge (air + fuel) flow (lb/sec)	Fuel-air ratio	Carburetor-air temperature (°F)	Ignition timing (deg B.T.C.)	
							Exhaust	Intake
Engine power or charge flow	400-1180	2600	24-48	1.03-2.47	0.080	60	34	28
	275-1250	2600	21-54	.84-2.76	.095	85	34	28
	490-1500	3000	26-58	1.38-3.53	.095	100	34	28
	925-1860	3200	41-73	2.35-4.40	.095	-5	34	28
	860- 940	2000-3200	40	1.70-2.18	.080	60	34	28
	750- 873	1800-3200	40	1.54-2.14	.095	86	34	28
	430- 630	1400-3200	30	.88-1.60	.095	86	34	28
	370- 640	1200-3200	30	.73-1.57	.080	60	34	28
Engine speed	662- 825	2300-3200	32-36	1.72	0.080	17	34	28
	647- 812	2300-3200	34-40	1.72	.080	140	34	28
	662- 868	2000-3200	33-40	1.76	.095	17	34	28
	546- 742	2000-3200	31-40	1.58	.095	143	34	28
Fuel-air ratio	740- 790	2600	35	1.75	0.064-0.115	66	34	28
	731- 837	2500	34-36	1.84	.062- .109	120	34	28
	760- 960	3000	39-41	2.20	.064- .118	25	34	28
Ignition timing	577- 655	2600	30	1.53	0.095	40	14-54	8-48
	780- 922	3000	40	2.19	.095	19	16-52	10-46
Carburetor-air temperature	786- 794	2500	34-35	1.83	0.095	80-184	34	28
	555- 590	2600	27-30	1.39	.090	-20-170	34	28
	659- 675	2600	30-34	1.58	.095	-27-144	34	28
	798- 846	3000	36-40	2.04	.095	-43-146	34	28
Exhaust pressure	427- 882	2600	35	1.36-1.92	0.085	32	34	28
	540- 612	2600	28-35	1.43	.053	24	34	28
	598- 702	2600	30-40	1.50	.100	4	34	28
	620- 726	2600	30-42	1.59	.085	39	34	28
	944- 994	3000	40-47	2.28	.085	-2	34	28
	592- 682	3000	30-41	1.65	.085	19	34	28
Coolant conditions	1000	2600	44	2.21	0.092	75	34	28
	1000	2600	44	2.21	.093	80	34	28
	1800	3200	70	4.19	.095	0	34	28
	1800	3200	70	4.19	.095	0	34	28
	1800	3200	70	4.19	.095	0	34	28
	1800	3200	70	4.19	.095	0	34	28
	780,1150	2600,3000	35,50	1.78,2.71	.095	83	34	28
	780,1160	2500,3000	36,50	1.78,2.71	.095	83	34	28
	780,1160	2500,3000	35,50	1.78,2.71	.095	83	34	28
	780,1160	2500,3000	35,50	1.78,2.71	.095	83	34	28
	780,1160	2600,3000	36,50	1.78,2.71	.095	83	34	28
	785	2600	36	1.80	.095	83	34	28
	795	2600	37	1.84	.095	84	34	28
	1800	3200	70	4.19	.095	0	34	28
	1000,1250, 1450	3000	42,51,60	2.39,2.94, 3.40	.095	50	34	28
	310-1200	2000-3000	24-53	.73-2.80	.054- .096	75	34	28
Engine running time	660	2500	32	1.57	0.095	82	34	28

^aEngine equipped with aftercooler.
^bAN-E-2 ethylene glycol.
^cCoolant-outlet temperature.

OF TEST CONDITIONS

Exhaust pressure (in. Hg abs.)	Coolant, glycol-water (percent by volume)		Coolant flow (gal/min)	Average coolant temperature (°F)	Engine coolant-outlet pressure (lb/sq in. gage)	Oil-inlet temperature (°F)	Engine
	Ethylene glycol	Water					
29-30	30	70	255	245	33	178	C
	30	70	250	245	35	180	D
	30	70	250	245	13-33	180	B[a]
	30	70	300	245	35	180	D
	30	70	255	245	33	176	C
	30	70	250	245	35	180	D
	30	70	250	245	35	180	D
	30	70	255	245	30	176	C
29-30	30	70	300	245	35	180	D
	30	70	300	245	35	180	D
	30	70	300	245	35	180	D
	30	70	300	245	35	180	D
29-30	30	70	270	245	35	180	C
	30	70	220	245	27	180	B[a]
	30	70	250	245	35	180	D
29-30	30	70	300	245	35	180	D
	30	70	300	245	35	180	D
29-30	30	70	230	245	25	180	B[a]
	30	70	270	245	30	180	C
	30	70	300	245	35	180	D
	30	70	300	245	35	180	D
10-61	30	70	250	245	35	180	D
5-45	30	70	250	245	35	180	D
10-60	30	70	250	245	35	180	D
10-61	30	70	250	245	35	180	D
12-52	30	70	250	245	35	180	D
12-59	30	70	250	245	35	180	D
29-30	[b]97	3	200-300	[c]220,[c]250,[c]275	10	180	A
	30	70	100-280	[c]220,[c]250	10,20,30	180	A
	30	70	50-300	[c]250,[c]275	15,20,25,35; 30,35,45	188	C
	50	50	55-300	[c]250,[c]275	15,25,35; 25,35,45	180	C
	30	70	55-225	223	18	180	E
	30	70	90-200	250	22	180	E
	0	100	50-300	245	35	180	D
	30	70	50-300	245	35	180	D
	50	50	50-300	245	35	180	D
	70	30	50-300	245	35	180	D
	[b]97	3	50-320	245	35	180	D
	30	70	200	149-283	35	180	D
	[b]97	3	200	150-302	35	180	D
	30	70	50,200	225	15-45	180	E
	30	70	220	[c]250,[c]275	20,25,35,38	170-226	B
	[b]97	3	144-255	[c]250	Varied with coolant flow	180	A
29-30	30	70	300	245	35	180	D

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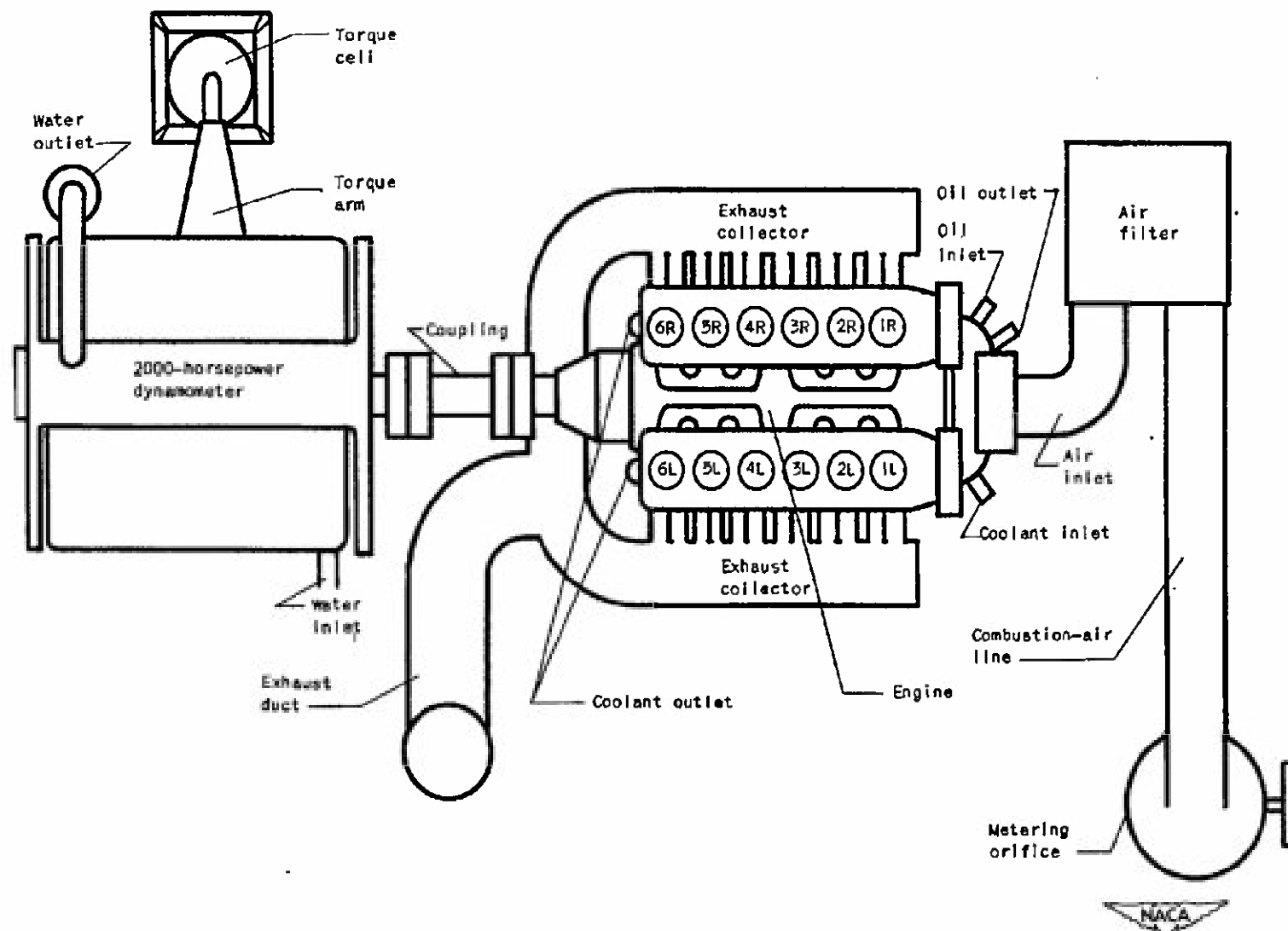
Figure I. - Plan view of engine setup.
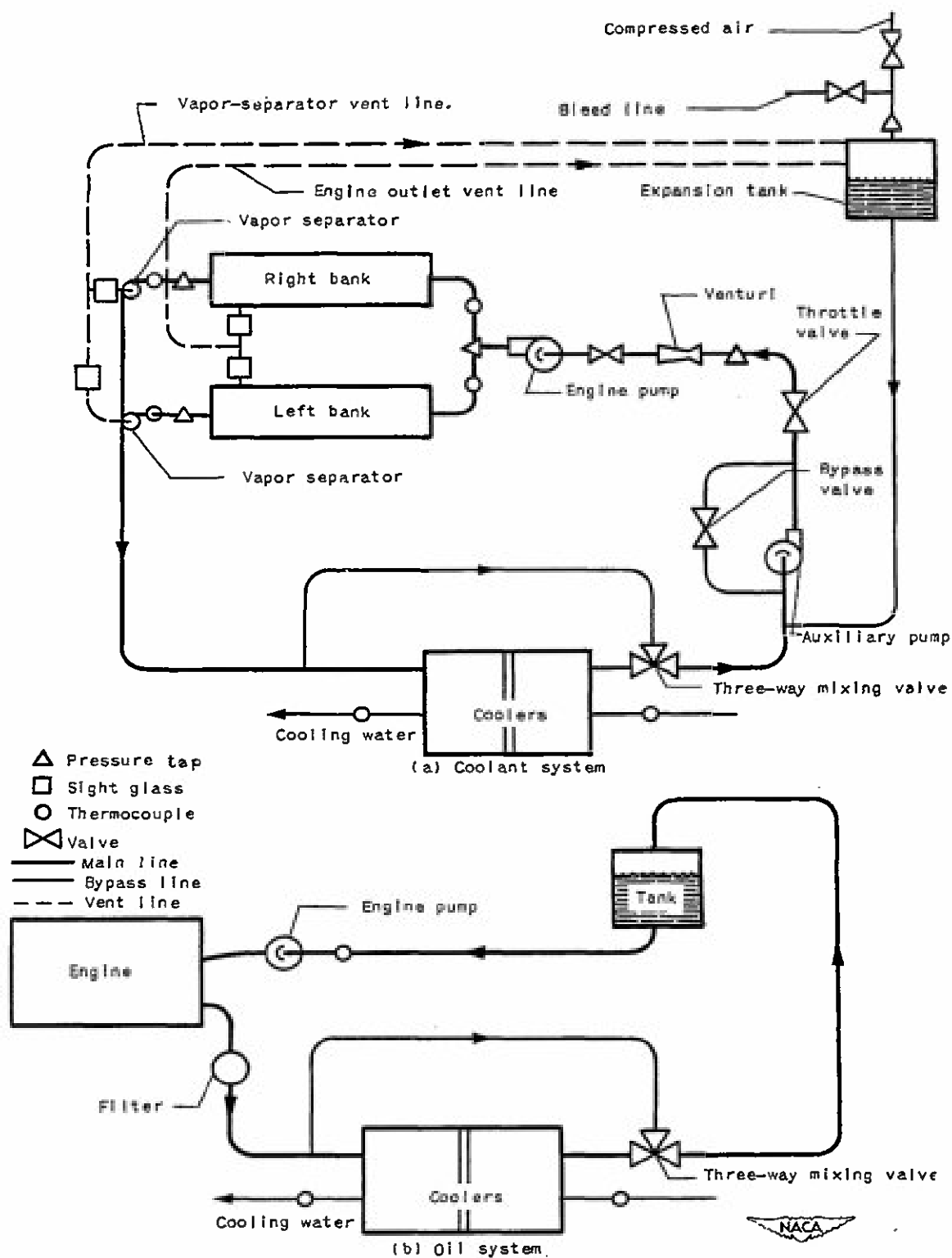
Figure 2. – Diagrammatic sketch of coolant and oil systems.
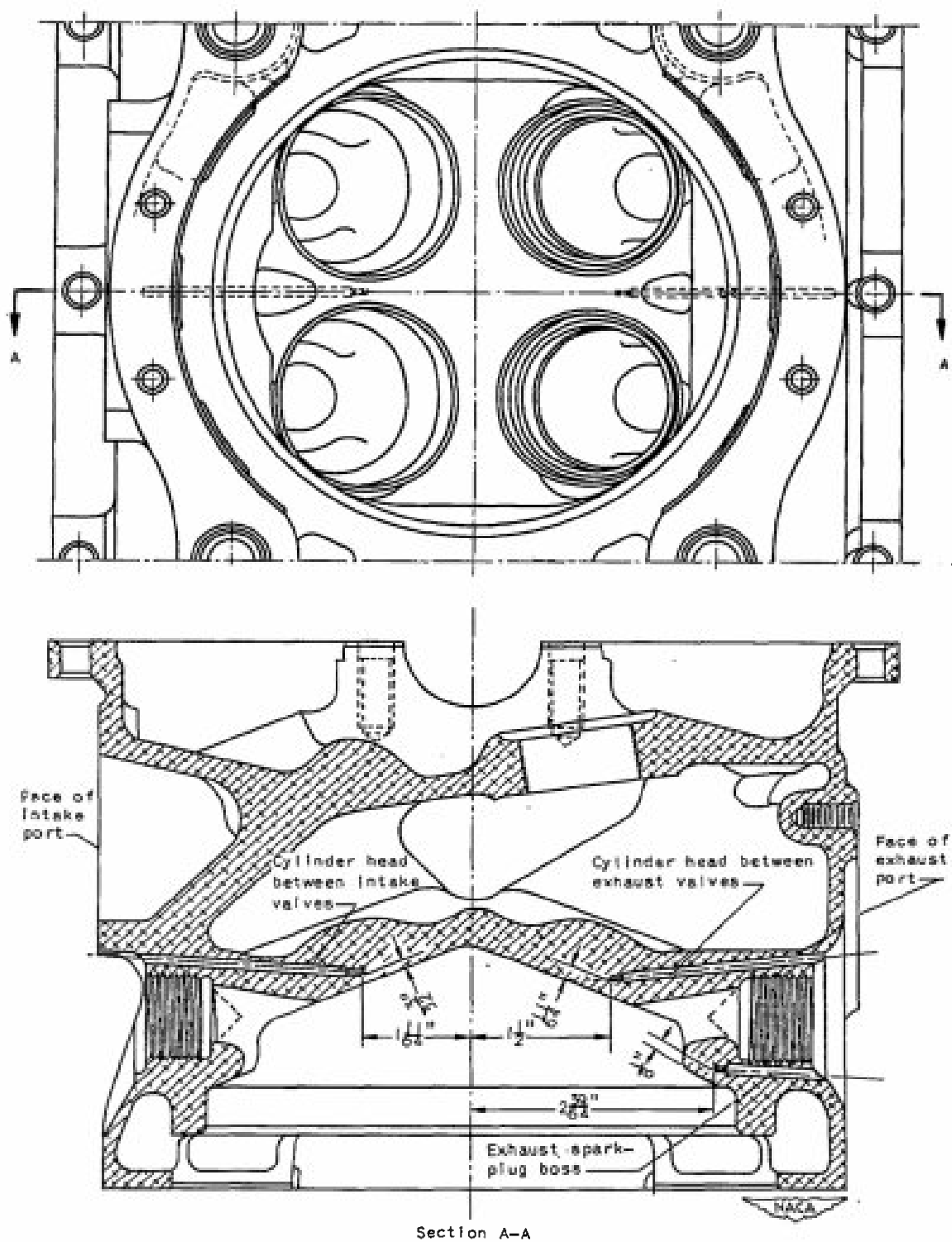
Figure 3. — Installation of cylinder-head thermocouples

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Exhaust-valve-guide
thermocouple

Cylinder-
barrel
thermocouple
leads

Pressure
fitting
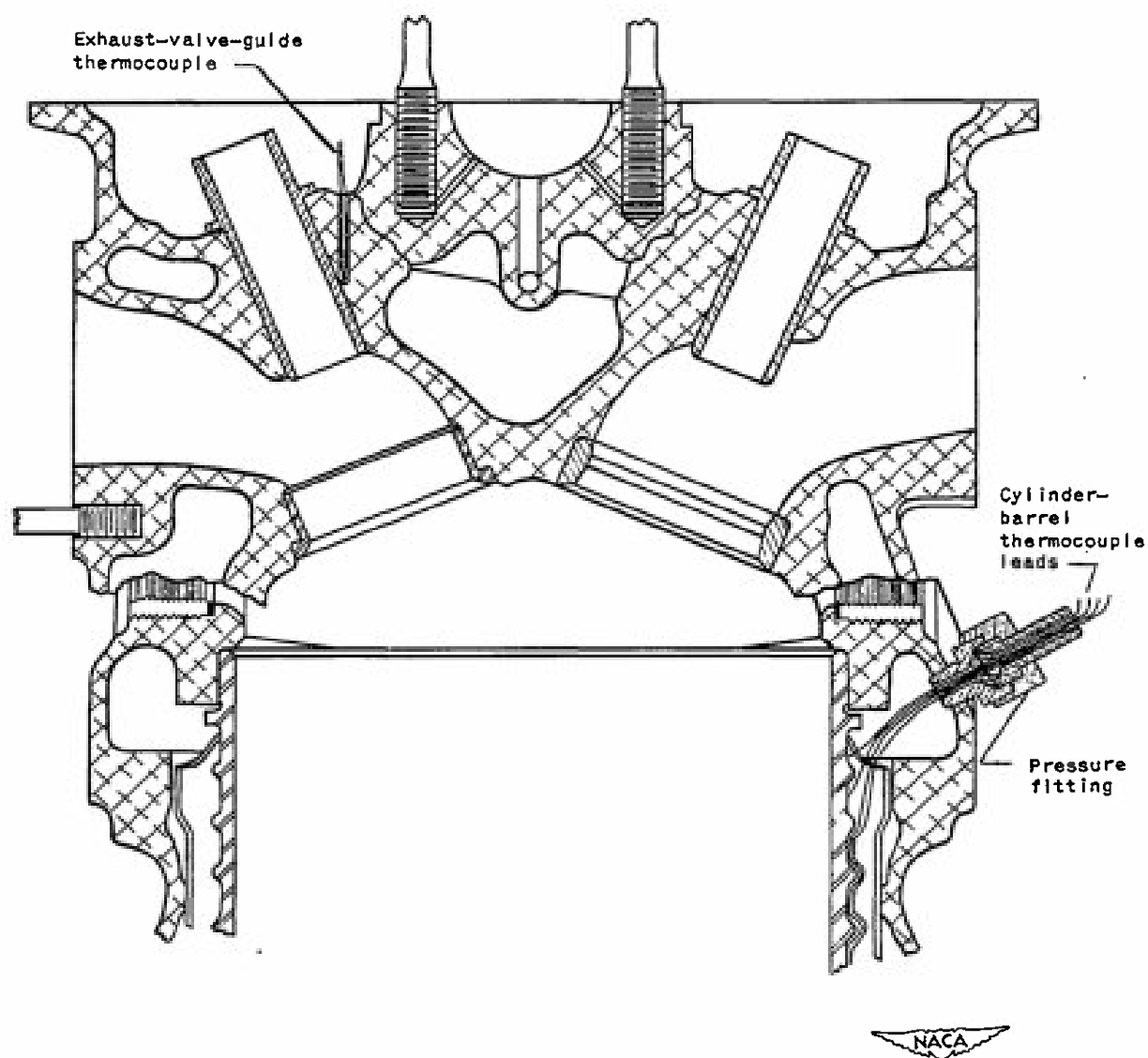
Figure 4. - Installation of exhaust-valve-guide thermocouple on engine
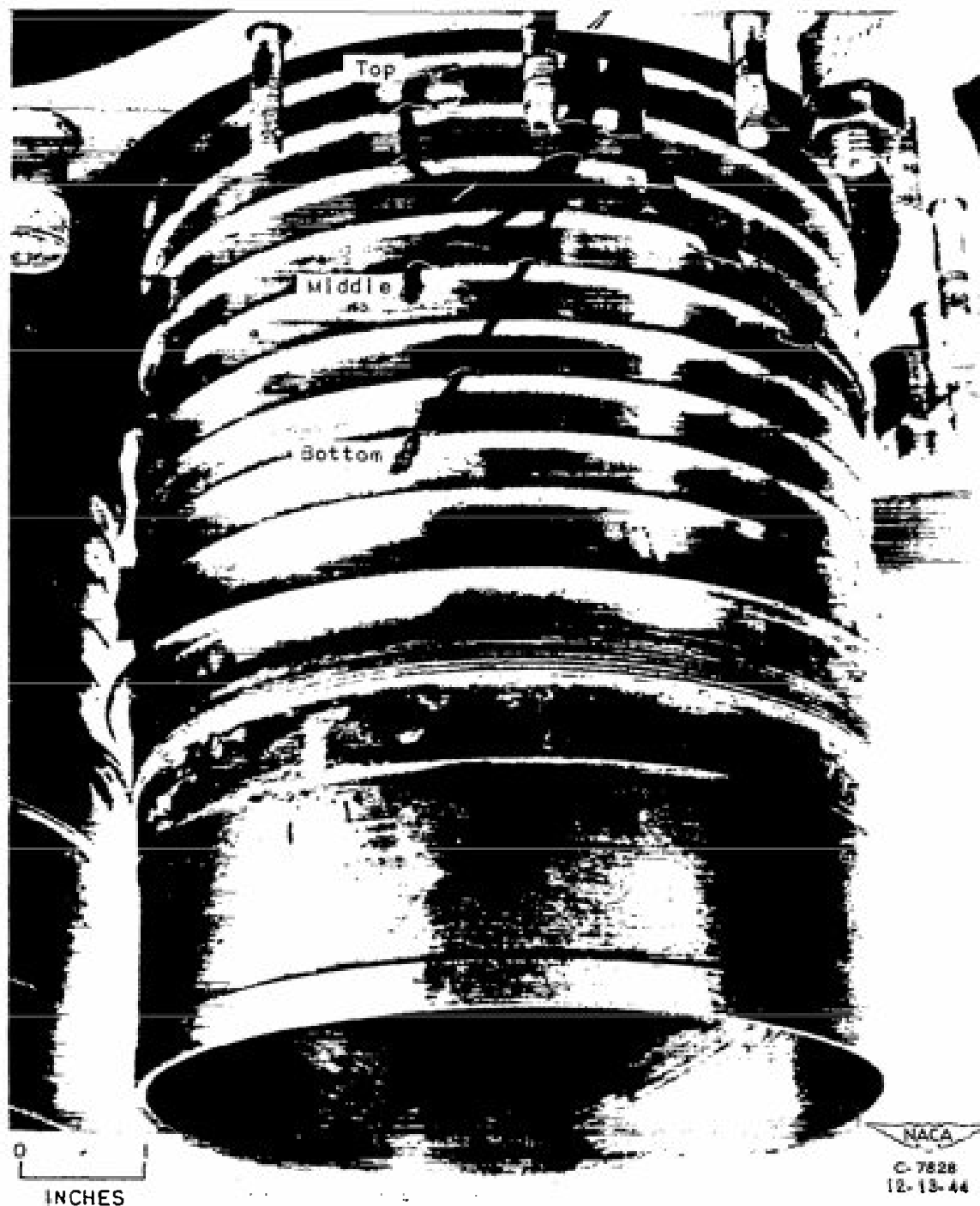and method of bringing out cylinder-barrel thermocouple leads.

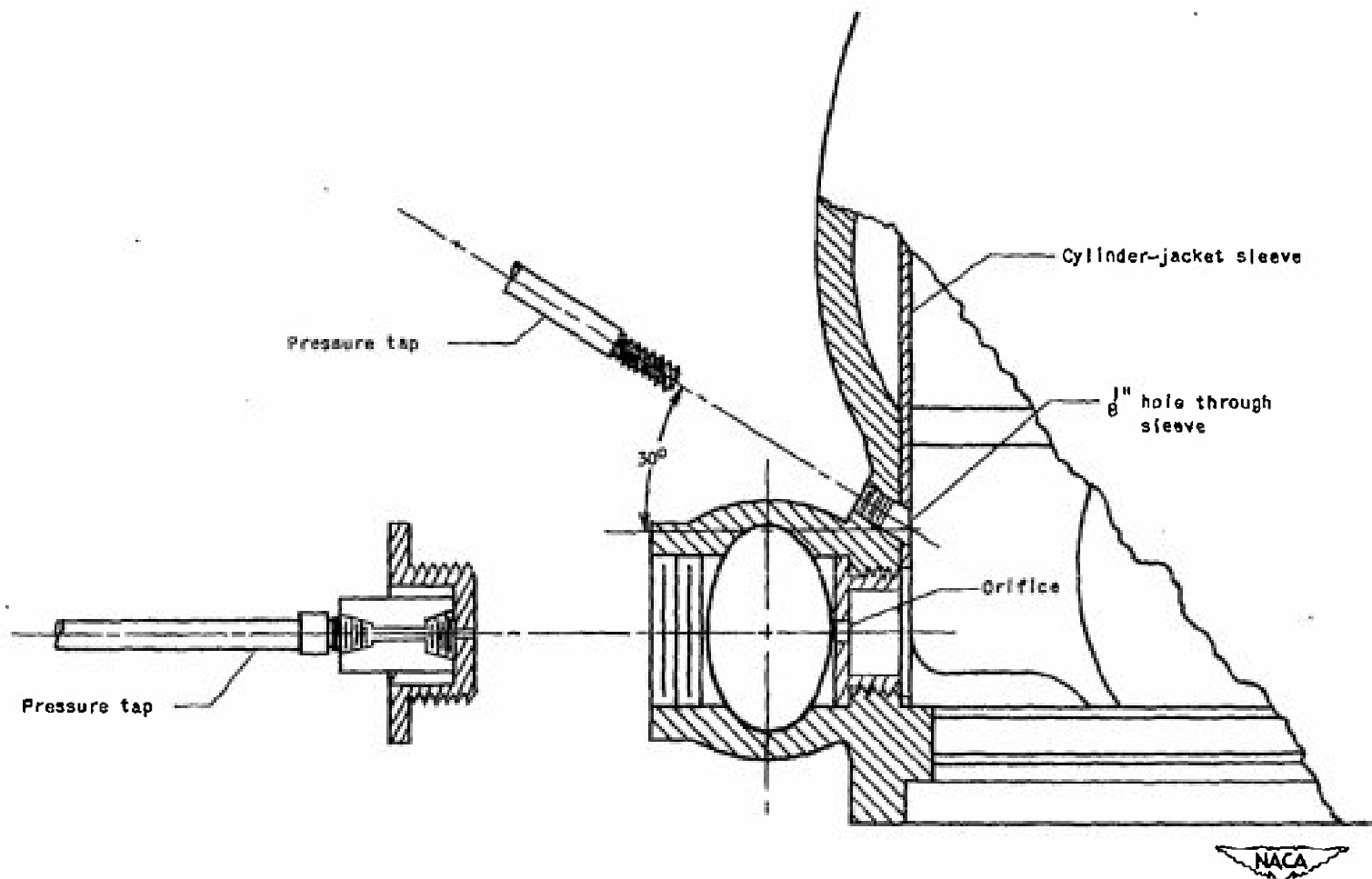Figure 5. — Installation of thermocouples on exhaust side of barrel
(Photograph by General Motors Corporation Allison Division.)

Figure 6. — Pressure-tap installation across orifice in cylinder jacket.

Figure 7. — Installation of pressure taps on engine.
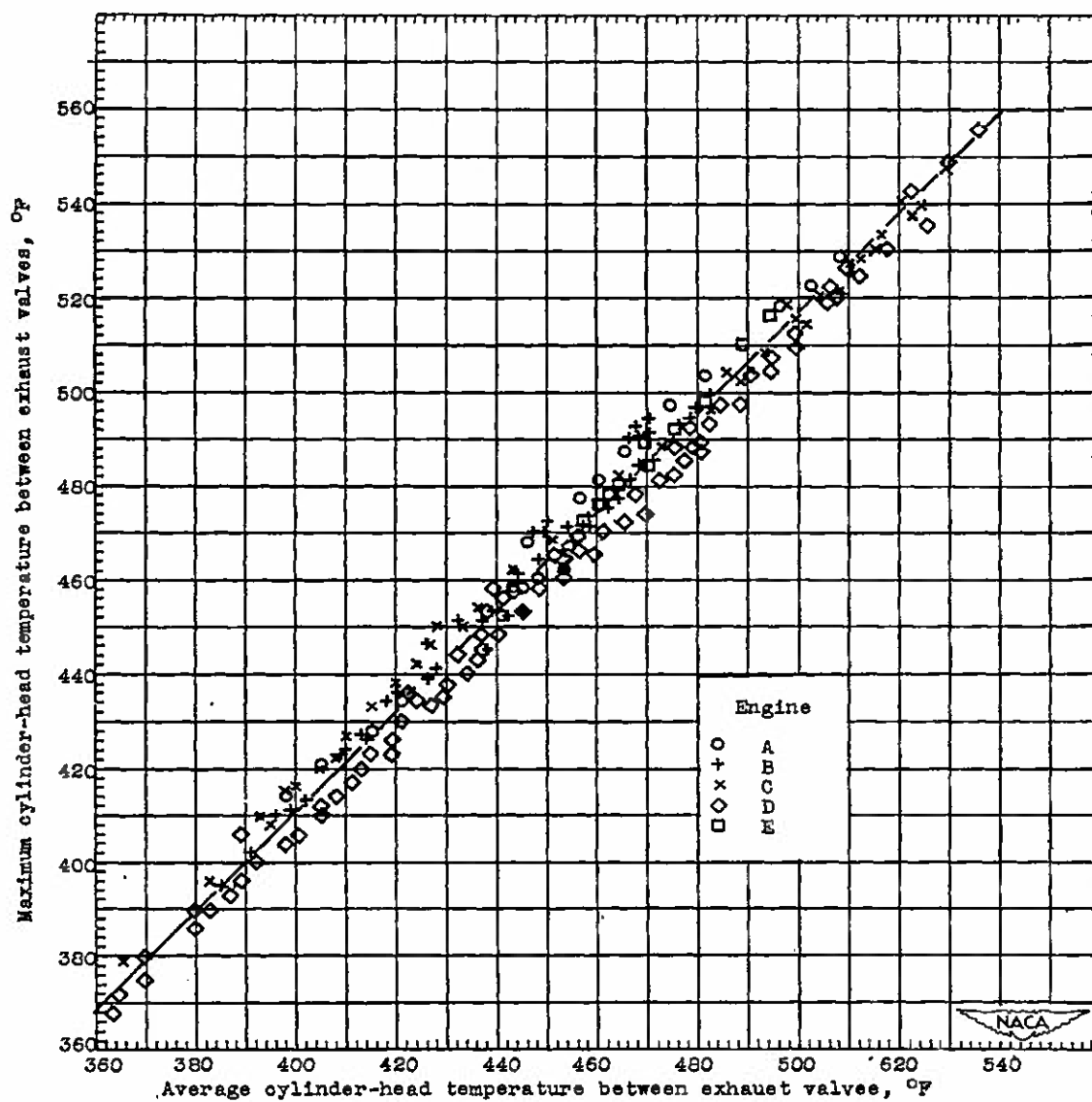
Figure 8. - Relation between average and maximum temperatures of cylinder head
between exhaust valves.

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NACA TN No. 1606
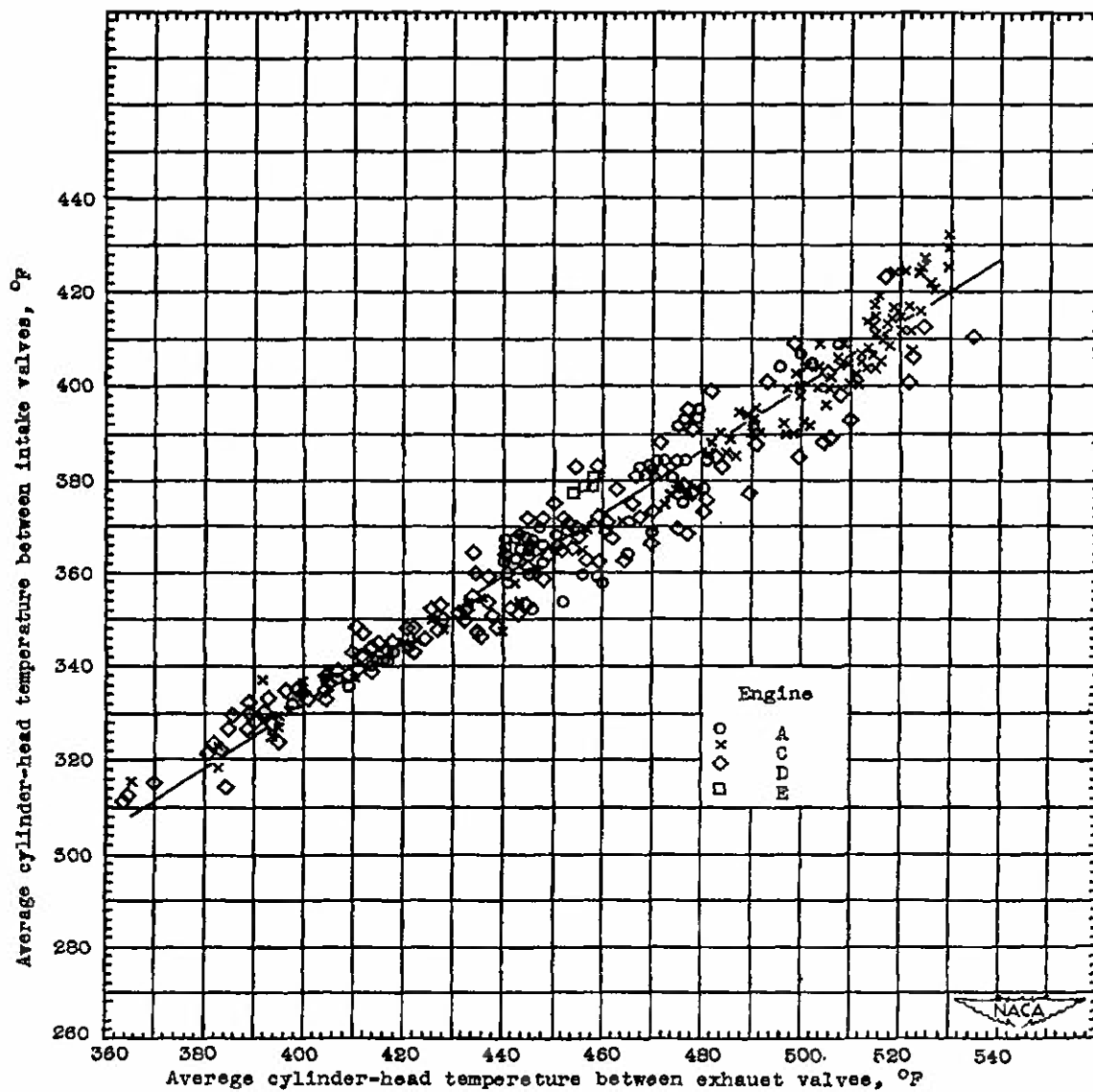
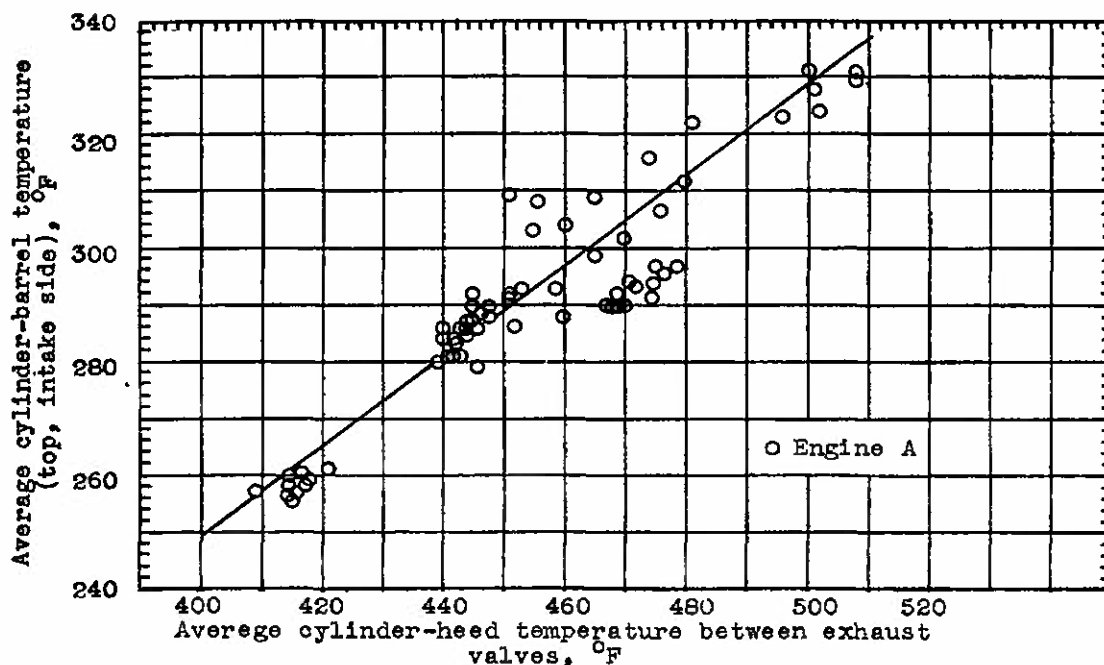Figure 9. - Relation between average cylinder-head temperature between exhaust valves and average cylinder-head temperature between intake valves.
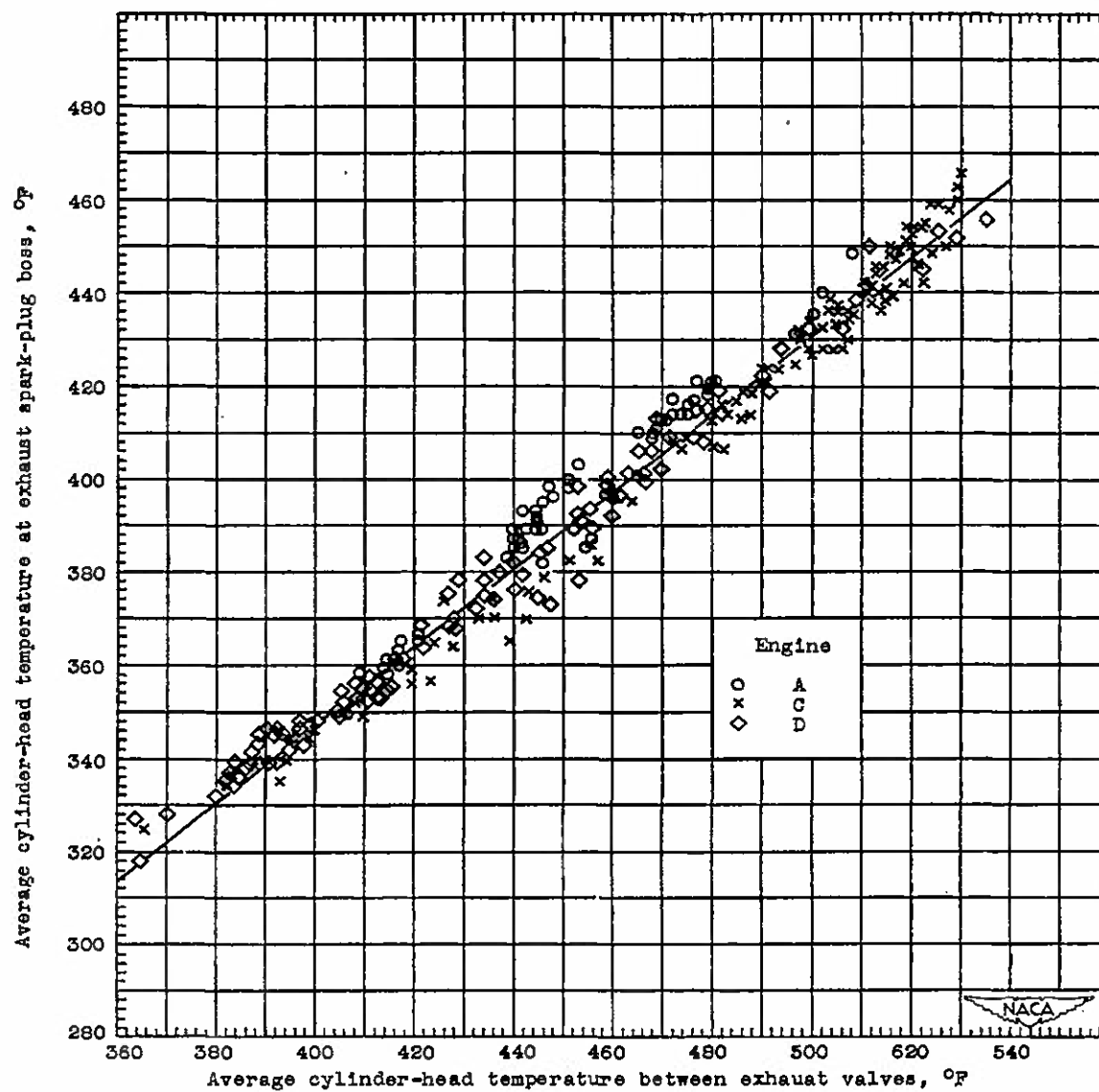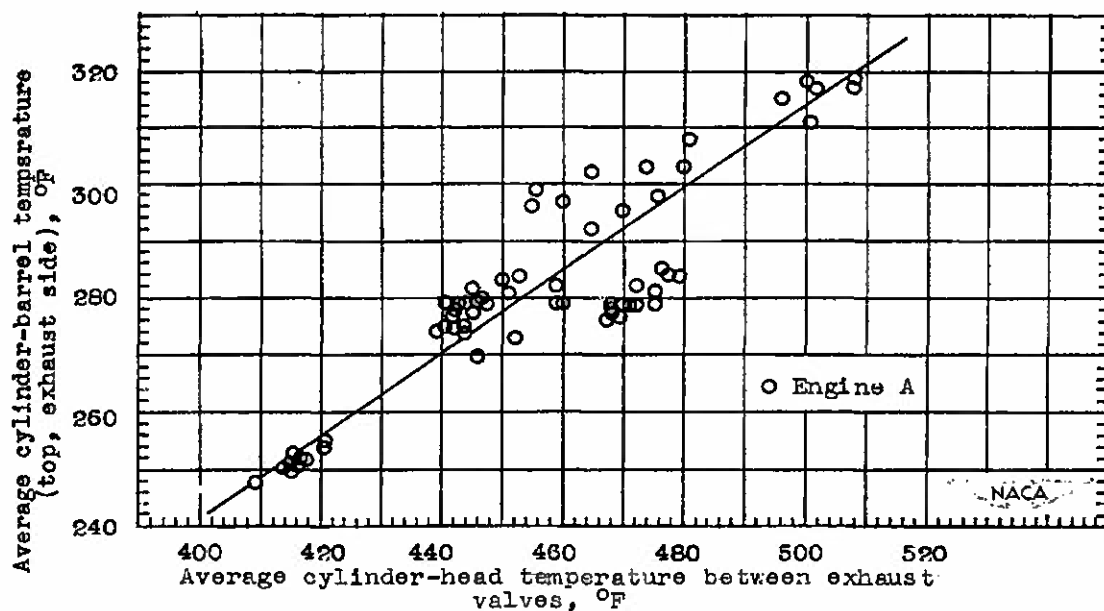
Figure 10. - Relation between average cylinder-head temperature between exhaust valves and average cylinder-head temperature at exhaust spark-plug boss.
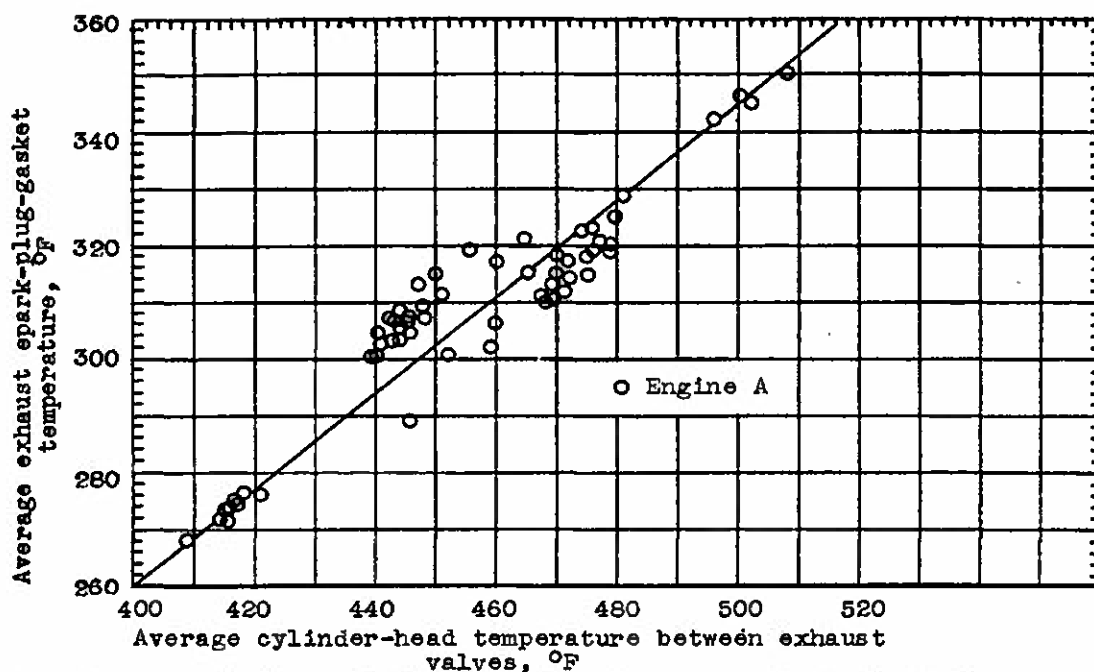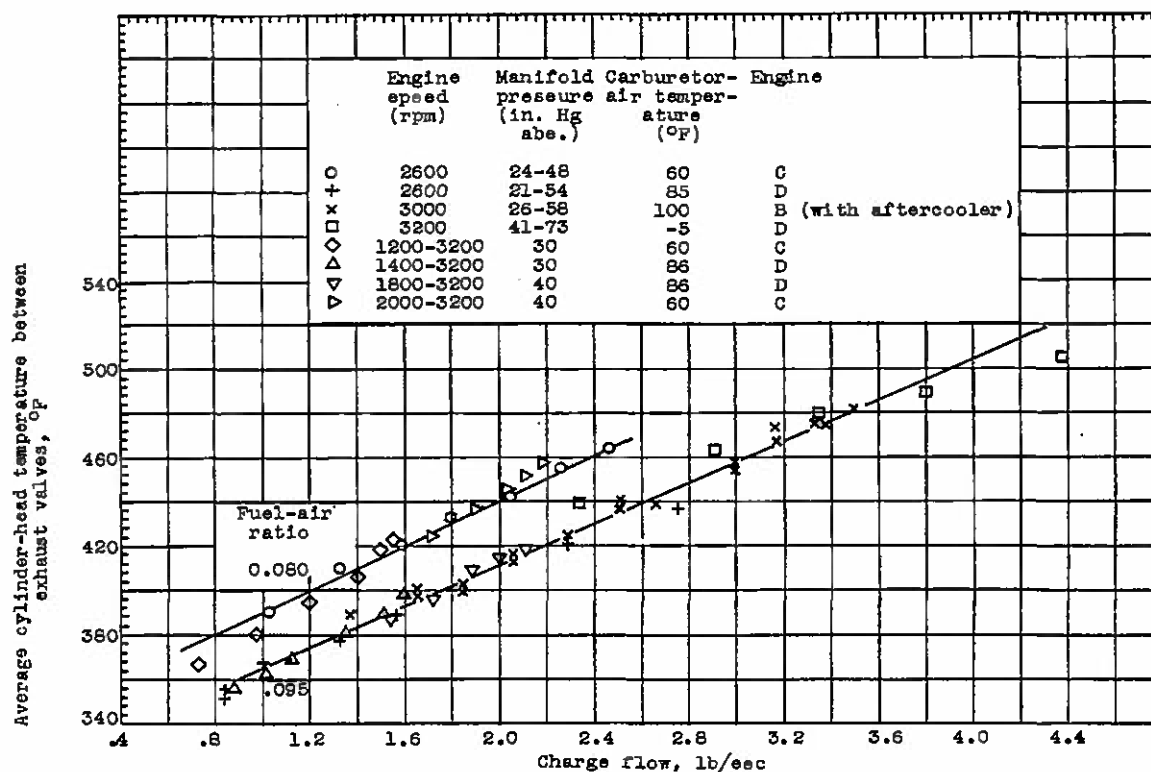
Figure 11. - Relation between average cylinder-head temperature between exhaust valves and average exhaust spark-plug-gasket temperature.
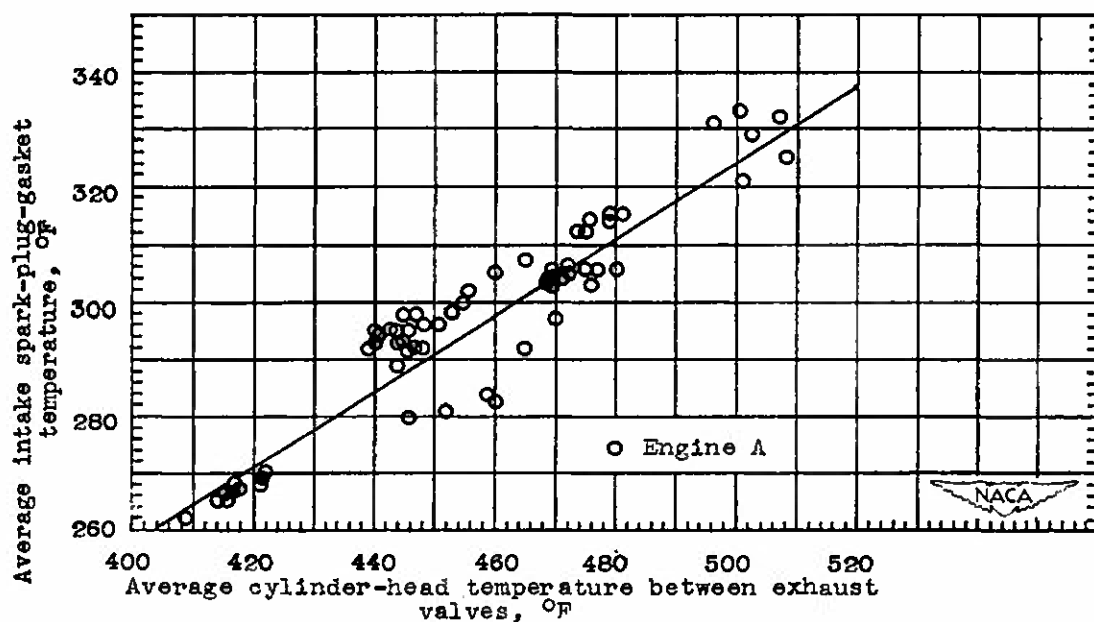
Figure 12. - Relation between average cylinder-head temperature between exhauet valvee and average intake spark-plug-gaeket temperature.
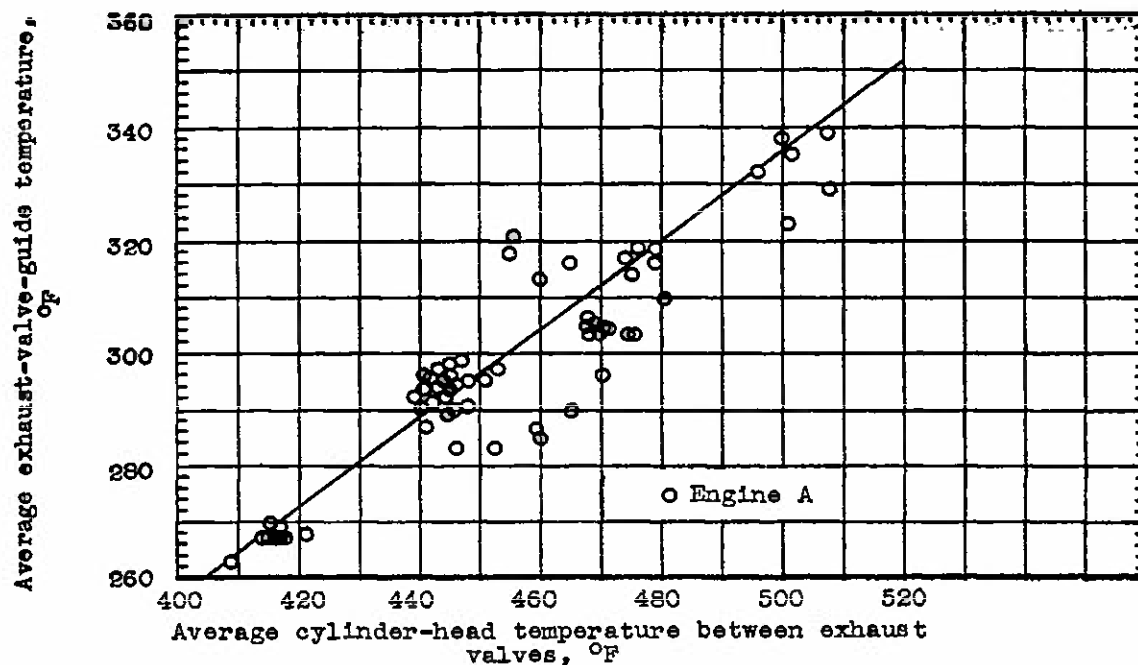
Figure 13. - Relation between average cylinder-head temperature between exhaust valves and average exhaust-valve-guide temperature.
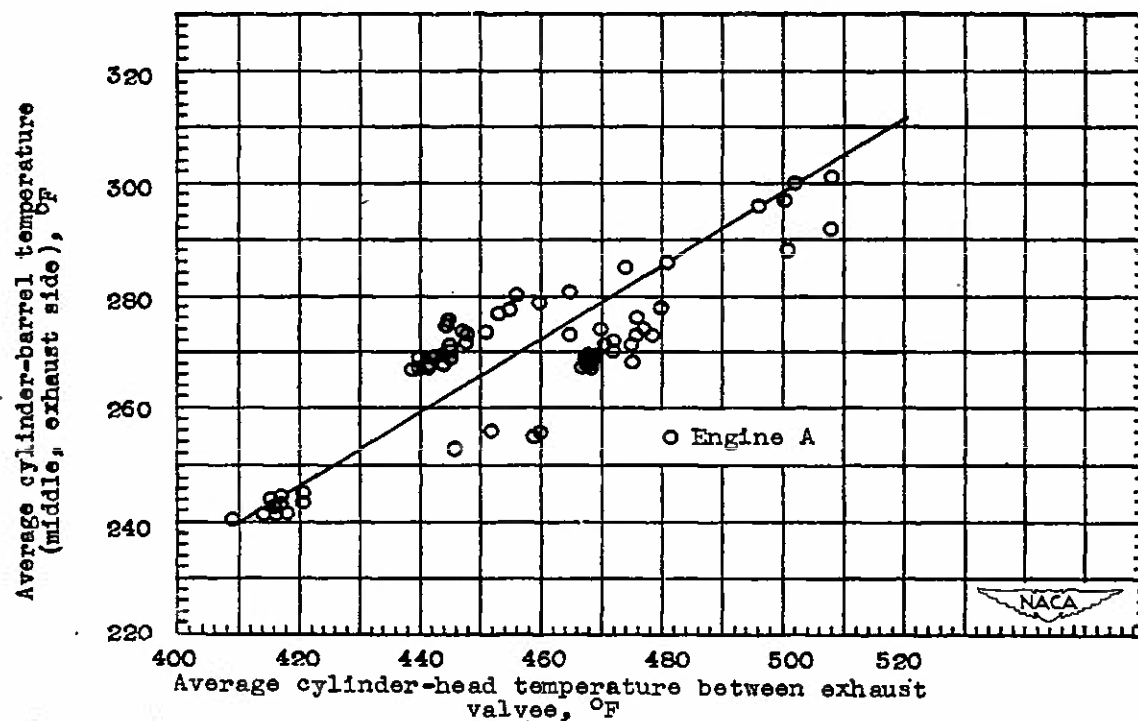
Figure 14. - Relation between average cylinder-head temperature between exhaust valves and average cylinder-barrel temperature (middle, exhaust side).

Figure 15. - Relation between average cylinder-head temperature between exhaust valves and average cylinder-barrel temperature (top, intake side).

Figure 16. - Relation between average cylinder-head temperature between exhaust valves and average cylinder-barrel temperature (top, exhaust side).

	Engine speed (rpm)	Manifold pressure (in. Hg abs.)	Carburetor-air temperature (°F)	Engine
○	2600	24-48	60	C
+	2600	21-54	85	D
×	3000	26-58	100	B (with aftercooler)
□	3200	41-73	-5	D
◇	1200-3200	30	60	C
△	1400-3200	30	86	D
▽	1800-3200	40	86	D
▷	2000-3200	40	60	C

Figure 17. - Effect of charge flow on average cylinder-head temperature between exhaust valves. Coolant, 30-70 ethylene glycol-water; average coolant temperature, 245° F; coolant flow, 250 to 300 gallons per minute; engine coolant-outlet pressure, 13 to 35 pounds per square inch gage.
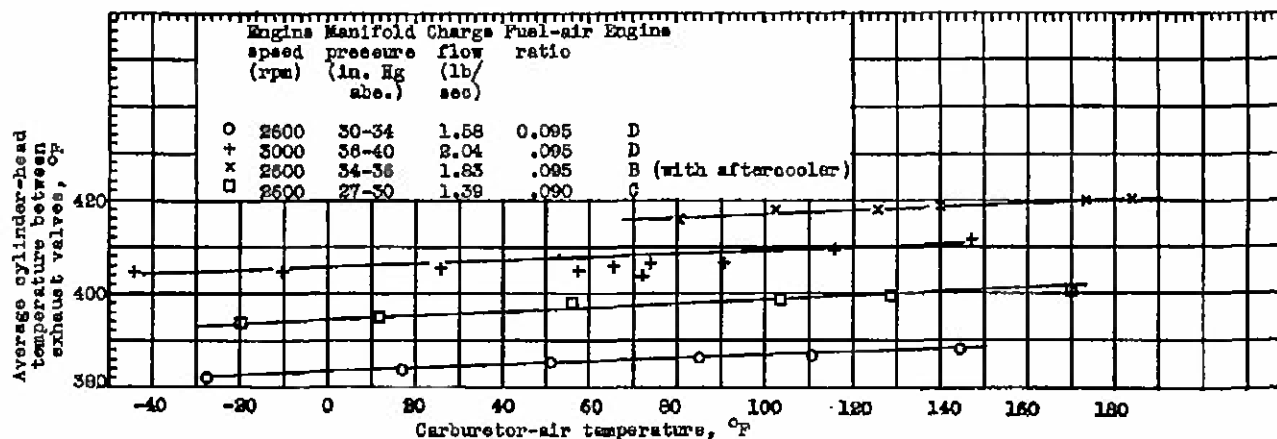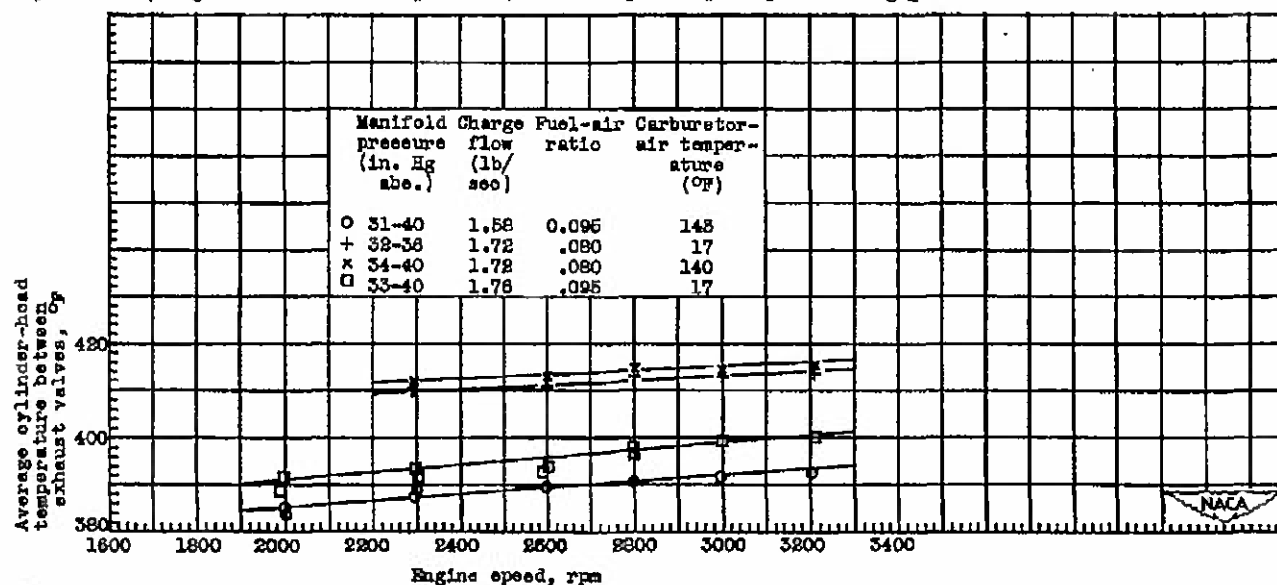

	Engine speed (rpm)	Manifold pressure (in. Hg abs.)	Fuel-air ratio	Carburetor-air temperature (°F)	Engine
○	2600	24-48	0.080	60	C
+	2600	21-54	.095	85	D
×	3000	26-58	.095	100	B (with aftercooler)
□	3200	41-73	.095	-5	D
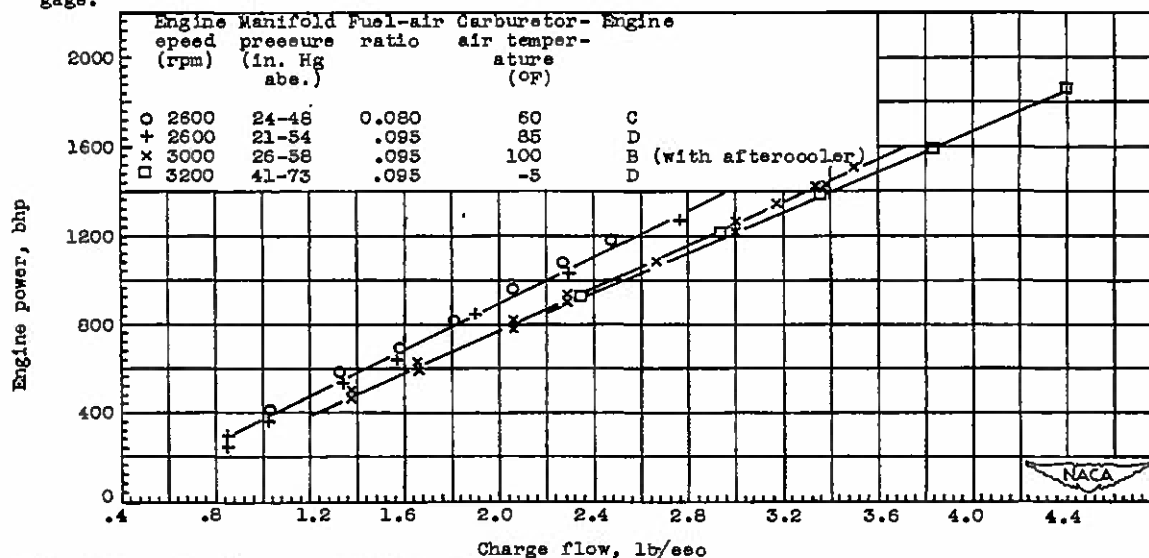
Figure 18. - Variation of engine power with charge flow. Coolant, 30-70 ethylene glycol-water; average coolant temperature, 245° F; coolant flow, 250 to 300 gallons per minute; engine coolant-outlet pressure, 13 to 35 pounds per square inch gage.

	Engine speed (rpm)	Manifold pressure (in. Hg abs.)	Charge flow (lb/sec)	Fuel-air ratio	Engine
O	2600	30-34	1.58	0.095	D
+	3000	36-40	2.04	.095	D
×	2600	34-36	1.83	.095	B (with aftercooler)
□	2600	27-30	1.39	.090	C

Figure 19. - Effect of carburetor-air temperature on average cylinder-head temperature between exhaust valves. Coolant, 30-70 ethylene glycol-water; average coolant temperature, 245° F; coolant flow, 250 to 300 gallons per minute; engine coolant-outlet pressure, 25 to 35 pounds per square inch gage.

	Manifold pressure (in. Hg abs.)	Charge flow (lb/sec)	Fuel-air ratio	Carburetor-air temperature (°F)
O	31-40	1.58	0.095	148
+	32-36	1.72	.080	17
×	34-40	1.72	.080	140
□	33-40	1.76	.095	17

Figure 20. - Effect of engine speed on average cylinder-head temperature between exhaust valves. Coolant, 30-70 ethylene glycol-water; average coolant temperature, 245° F; coolant flow, 300 gallons per minute; engine coolant-outlet pressure, 35 pounds per square inch gage; engine D.

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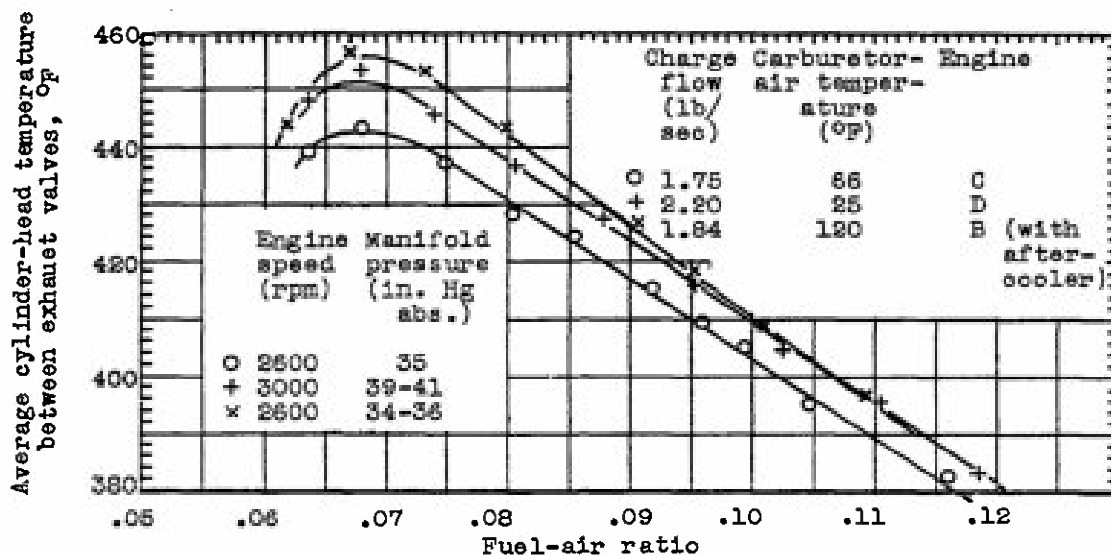
Figure 21. - Effect of fuel-air ratio on average cylinder-head temperature between exhaust valves. Coolant, 30-70 ethylene glycol-water; average coolant temperature, 245° F; coolant flow, 220 to 270 gallons per minute; engine coolant-outlet pressure, 27 to 35 pounds per square inch gage.
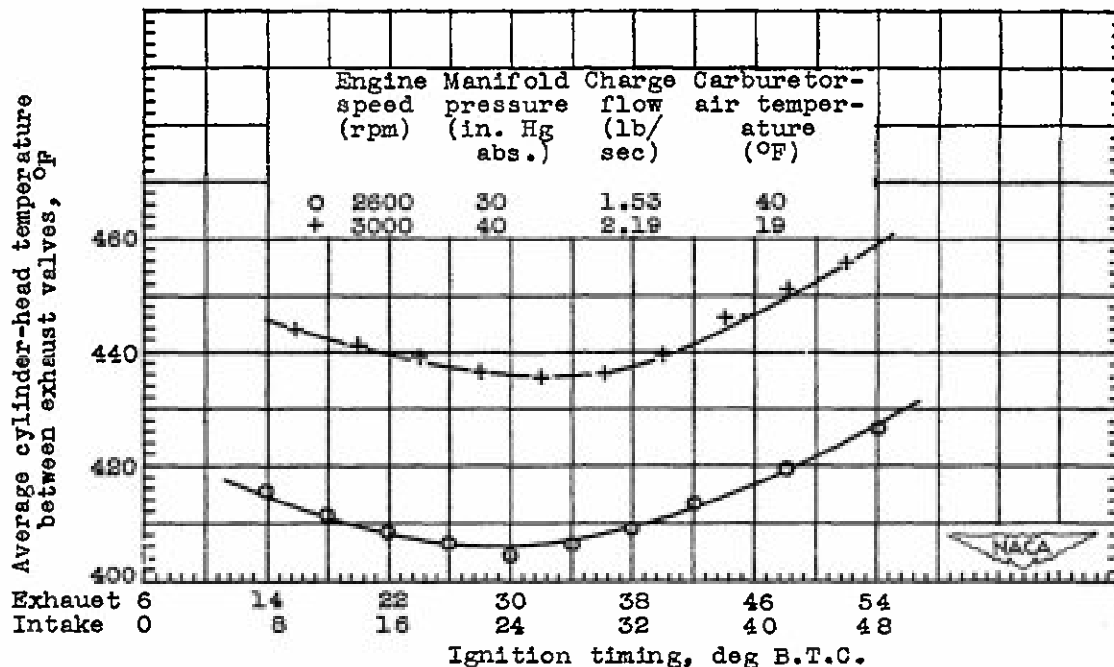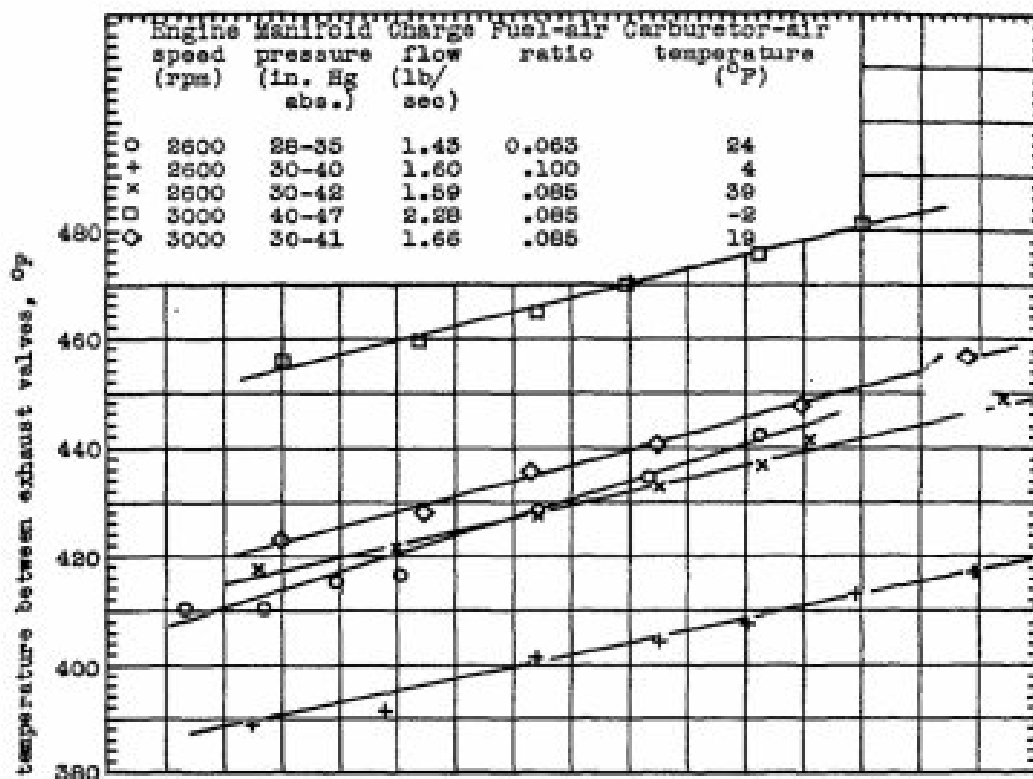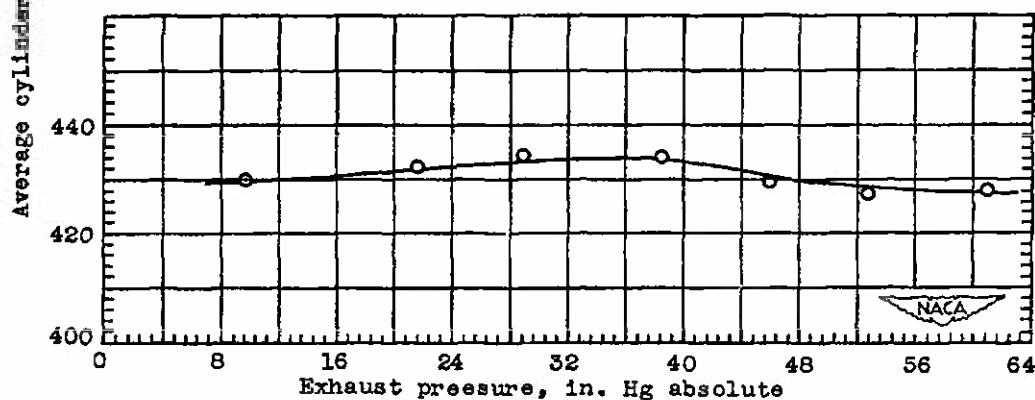

Figure 22. - Effect of ignition timing on cylinder-head temperature between exhaust valves. Fuel-air ratio, 0.095; coolant, 30-70 ethylene glycol-water; average coolant temperature, 245° F; coolant flow, 300 gallons per minute; engine coolant-outlet pressure, 35 pounds per square inch gage; engine D.
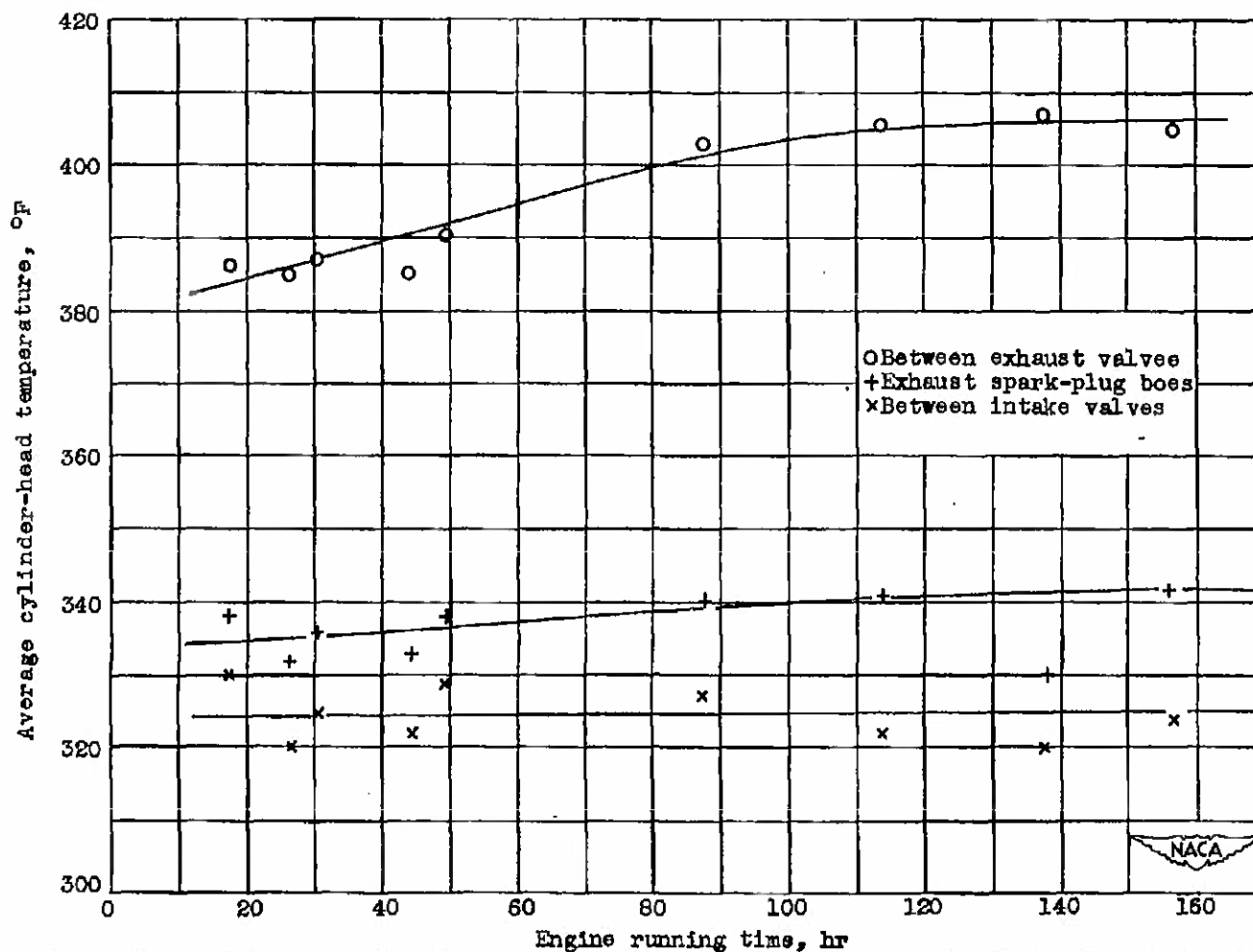
	Engine speed (rpm)	Manifold pressure (in. Hg abs.)	Charge flow (lb/ sec)	Fuel-air ratio	Carburetor-air temperature (°F)
O	2600	28-35	1.43	0.063	24
+	2600	30-40	1.60	.100	4
×	2600	30-42	1.59	.085	39
□	3000	40-47	2.28	.085	-2
◇	3000	30-41	1.66	.085	19

(a)   Constant charge flow with variable manifold pressure.

(b)   Constant manifold pressure with variable charge flow.
      Engine speed, 2600 rpm; manifold pressure, 35 inches
      mercury absolute; charge flow, 1.36 to 1.92 pounds
      per second; fuel-air ratio, 0.085; carburetor-air
      temperature, 32° F.

Figure 23. - Effect of exhaust pressure on average cylinder-head
   temperature between exhaust valves. Coolant, 30-70 ethylene glycol-
   water; average coolant temperature, 245° F; coolant flow, 250 gallons
   per minute; engine coolant-outlet pressure, 35 pounds per square
   inch gage; engine D.

Figure 24. - Variation of average cylinder-head temperatures with engine running time. Engine speed, 2600 rpm; manifold pressure, 32 inches mercury absolute; charge flow, 1.57 pounds per second; fuel-air ratio, 0.095; carburetor-air temperature, 82° F; coolant, 30-70 ethylene glycol-water; coolant flow, 300 gallons per minute; average coolant temperature, 245° F; engine coolant-outlet pressure, 35 pounds per square inch gage; engine D.
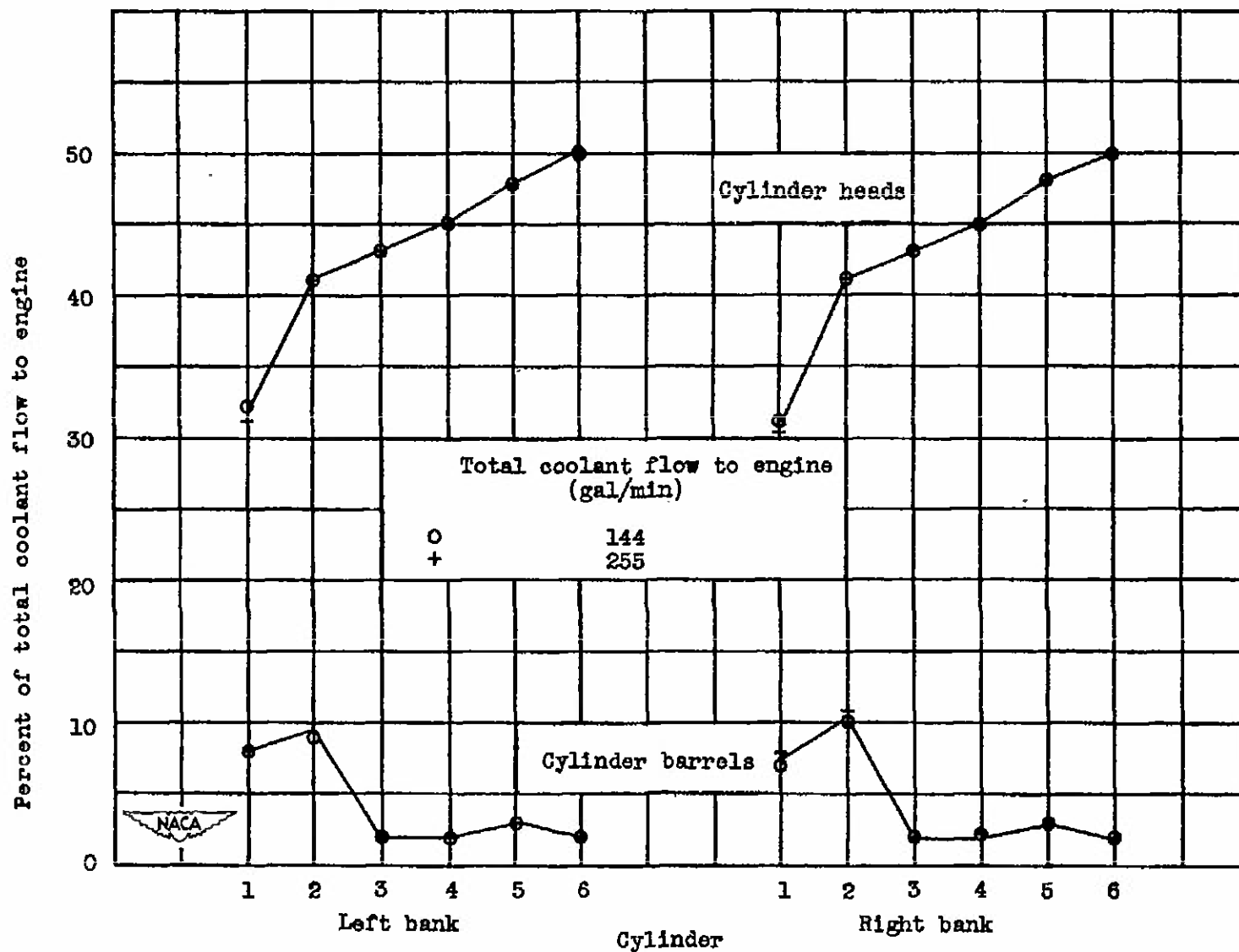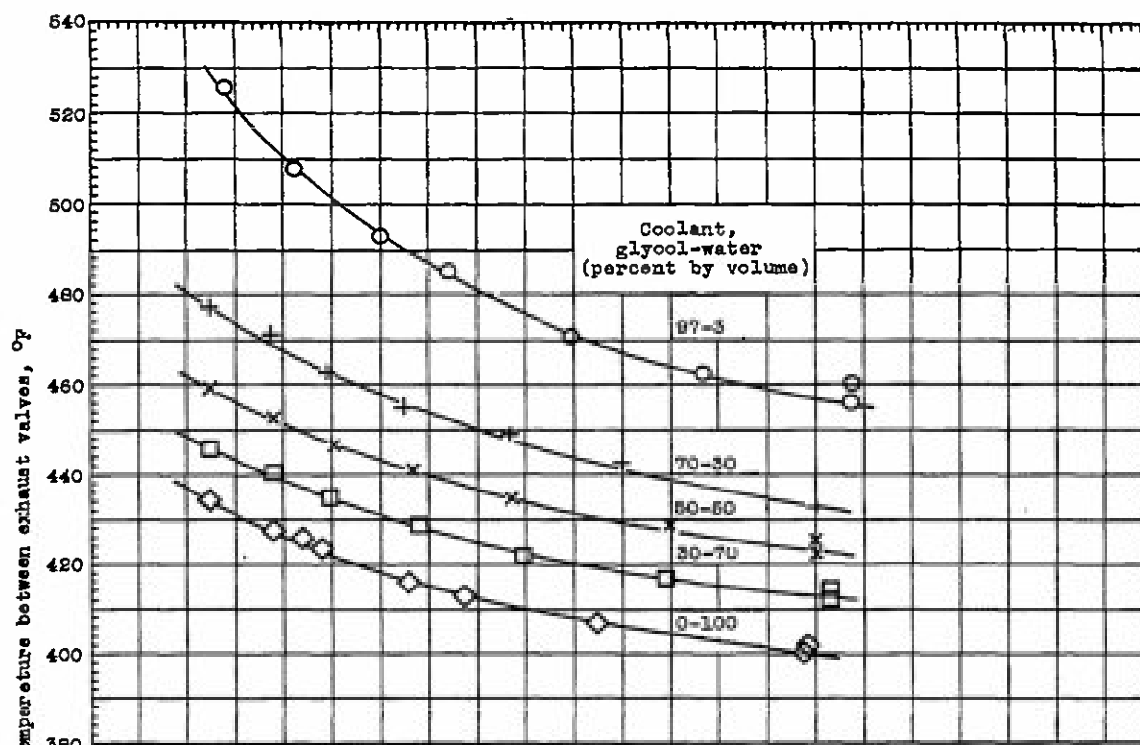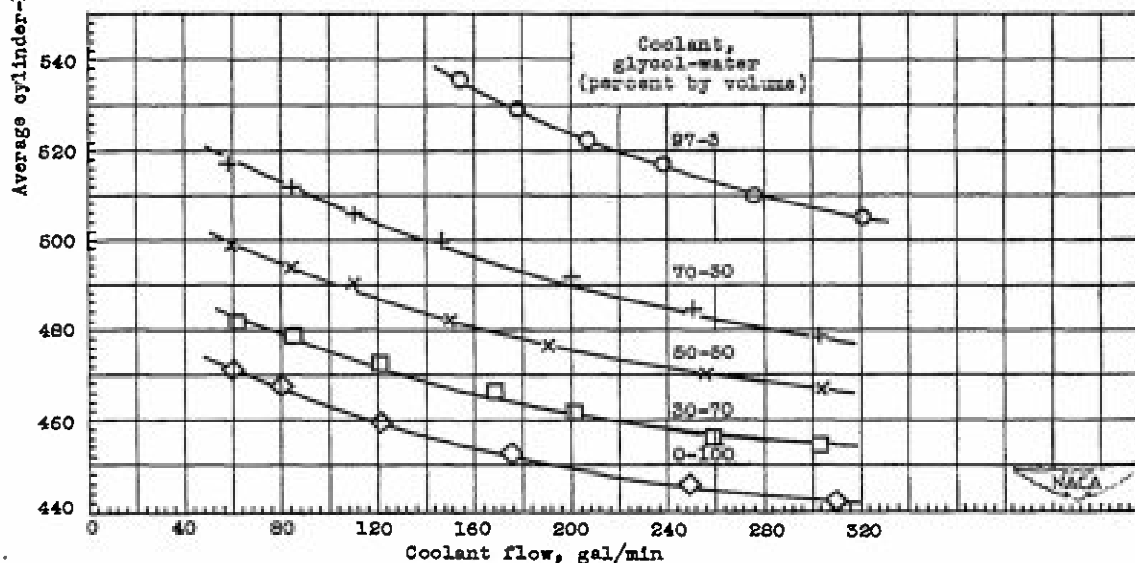
Figure 25. - Coolant-flow distribution in cylinder banks.

(a) Engine power, 780 brake horsepower; engine speed, 2600 rpm; manifold pressure, 36 inches mercury absolute; charge flow, 1.78 pounds per second.

(b) Engine power, 1160 brake horsepower; engine speed, 3000 rpm; manifold pressure, 50 inches mercury absolute; charge flow, 2.71 pounds per second.

Figure 26. - Effect of coolant flow on average cylinder-head temperature between exhaust valves for constant average coolant temperature and nonboiling coolant conditions with various ethylene glycol-water mixtures. Fuel-air ratio, 0.095; carburetor-air temperature, 83° F; average coolant temperature, 245° F; engine coolant-outlet pressure, 35 pounds per square inch gage; engine D.
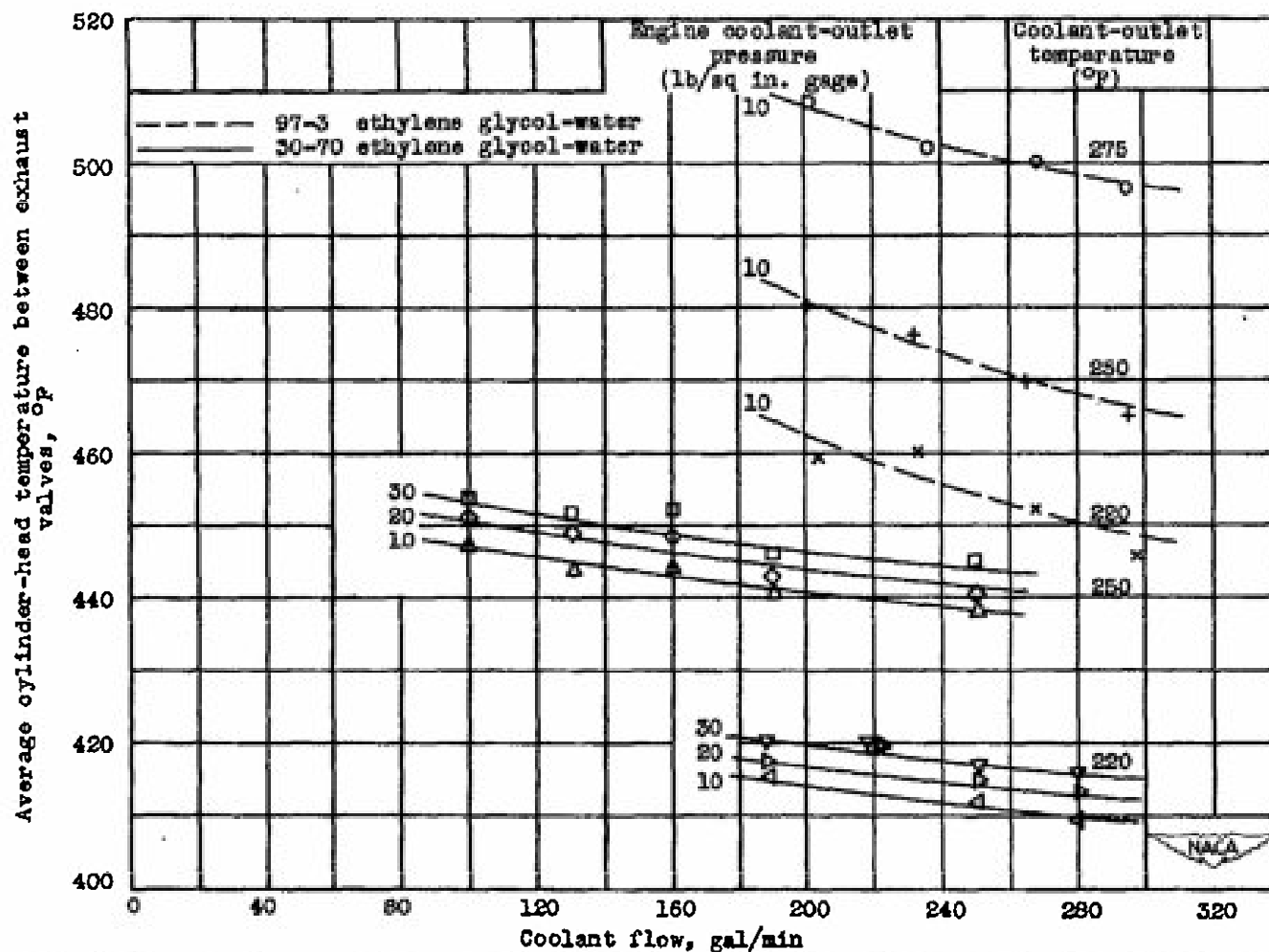
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Figure 27. - Effect of coolant flow on average cylinder-head temperature between exhaust valves for several constant coolant-outlet temperatures and nonboiling coolant conditions. Engine power, 1000 brake horsepower; engine speed, 2600 rpm; manifold pressure, 44 inches mercury absolute; charge flow, 2.21 pounds per second; fuel-air ratio, 0.092 to 0.093; carburetor-air temperature, 75° to 80° F; engine A.
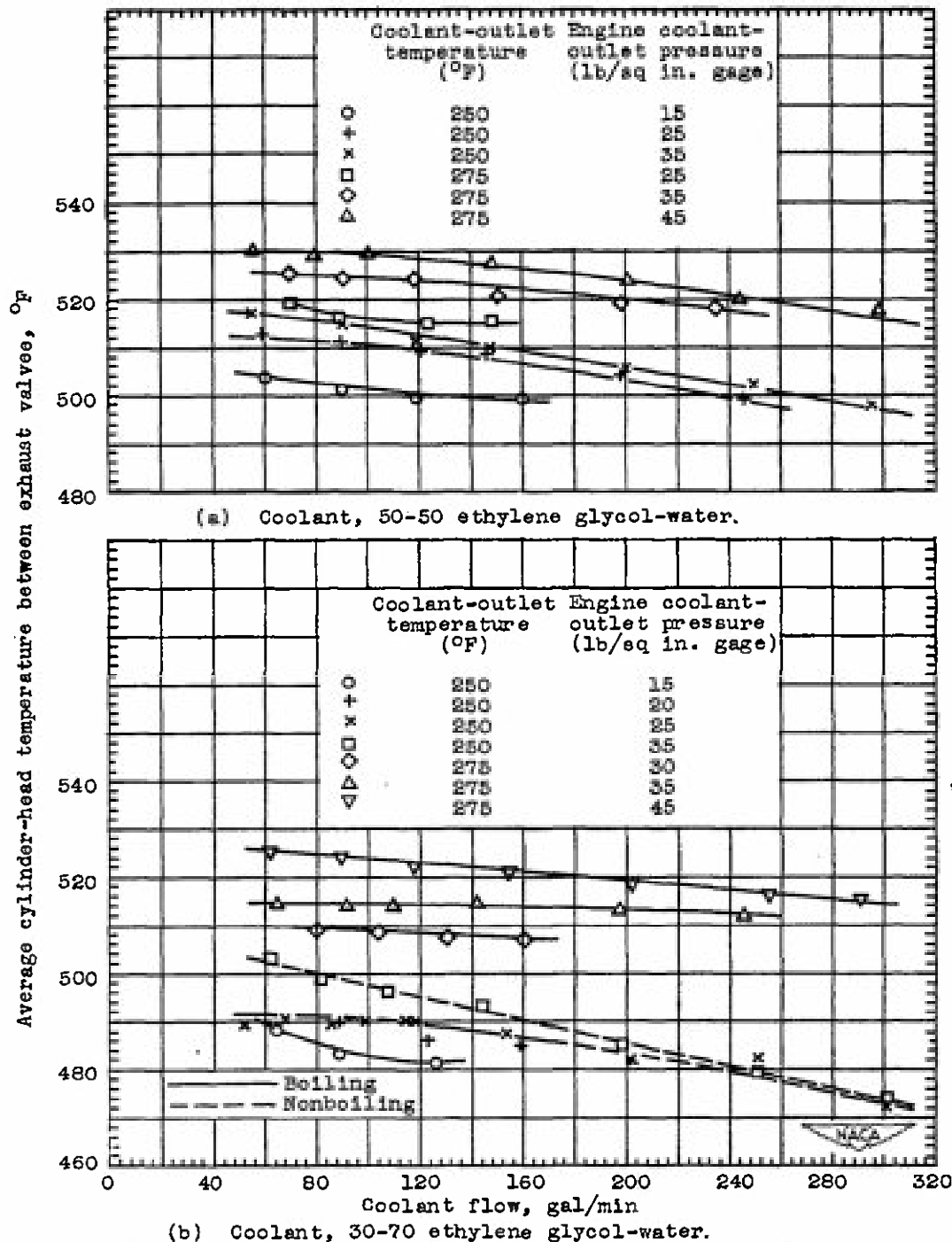
(a)   Coolant, 50-50 ethylene glycol-water.

(b)   Coolant, 30-70 ethylene glycol-water.

Figure 28. - Effect of coolant flow on average cylinder-head temperature
  between exhaust valves for several constant coolant-outlet temperatures
  and for boiling and nonboiling coolant conditions. Engine power, 1800
  brake horsepower; engine speed, 3200 rpm; manifold pressure, 70 inches
  mercury absolute; charge flow, 4.19 pounds per second; fuel-air ratio,
  0.095; carburetor-air temperature, 0° F; engine C.

(a)   Average coolant temperature, 223° F; engine coolant-
        outlet pressure, 18 pounds per square inch gage.

(b)   Average coolant temperature, 250° F; engine coolant-
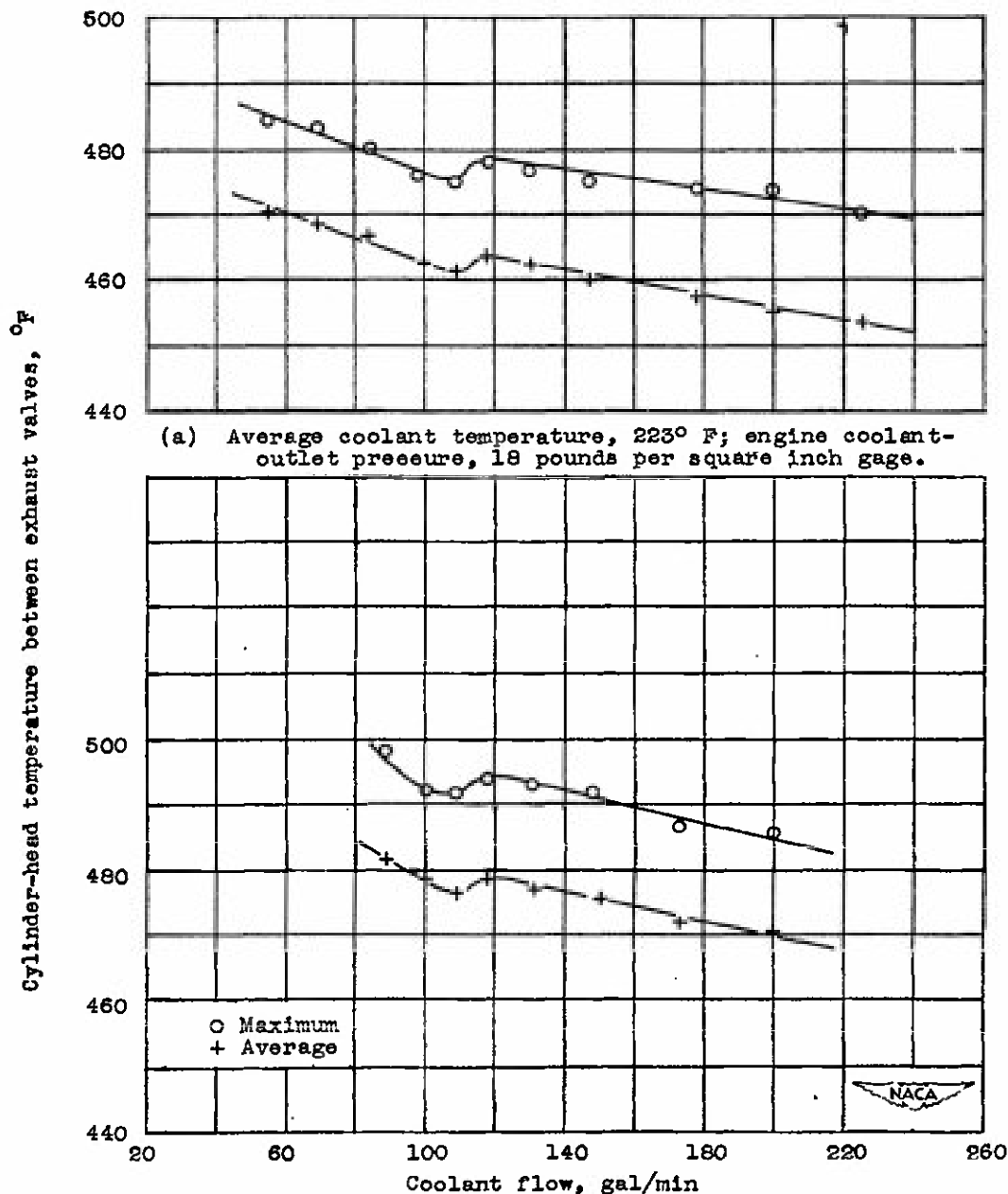   outlet pressure, 22 pounds per square inch gage.

Figure 29. - Effect of coolant flow on cylinder-head temperature
   between exhaust valves for several average coolant temperatures.
   Engine power, 1800 brake horsepower; engine speed, 3200 rpm;
   manifold pressure, 70 inches mercury absolute; charge flow,
   4.19 pounds per second; fuel-air ratio, 0.095, carburetor-air
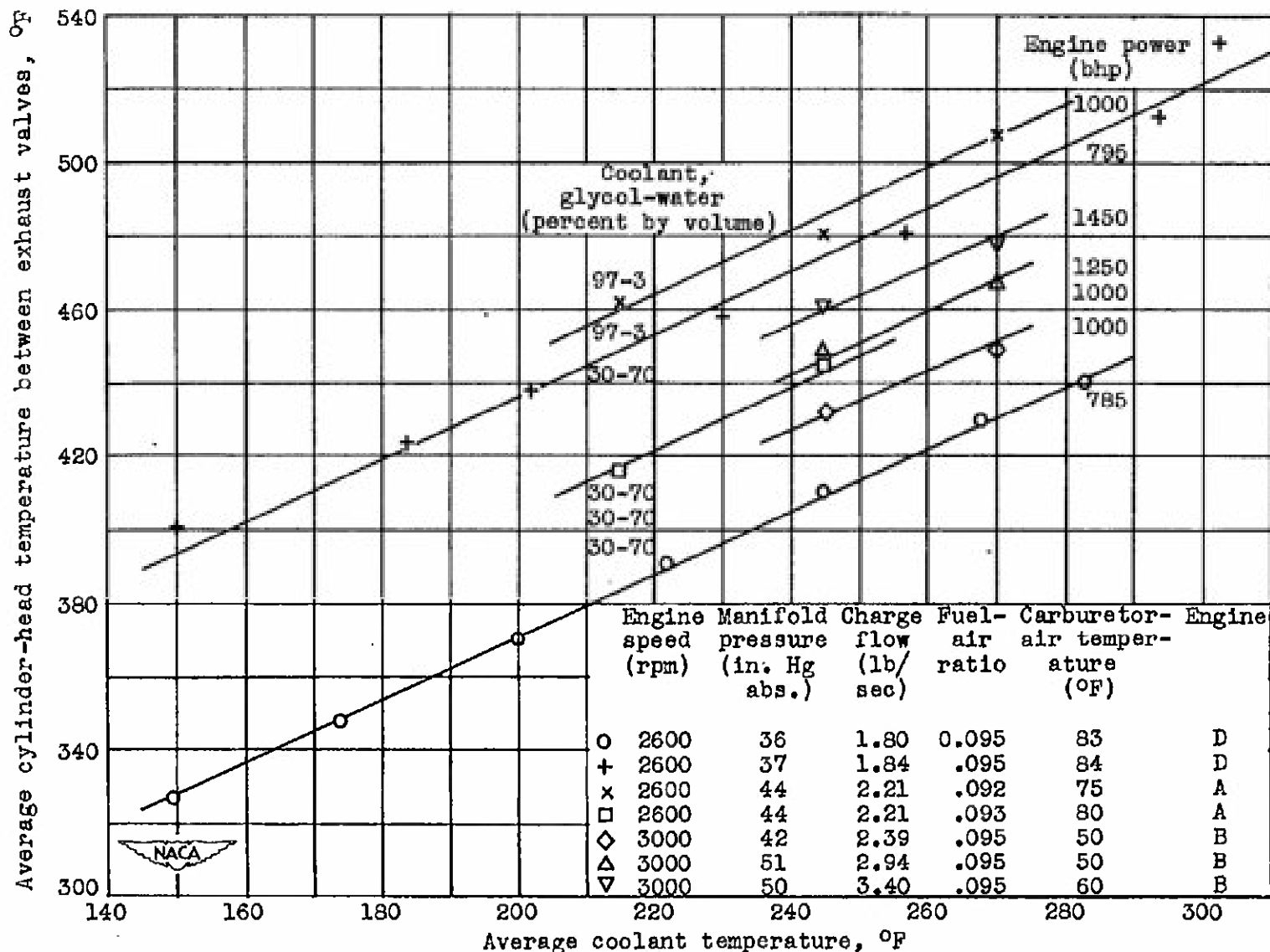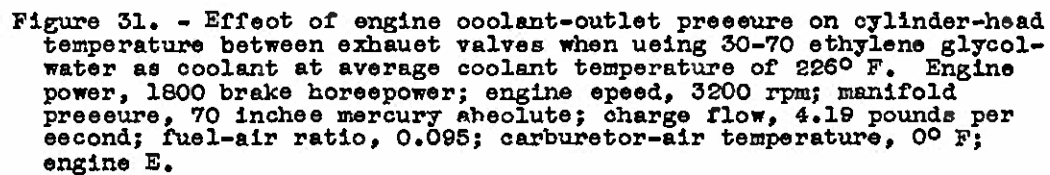   temperature, 0° F; coolant, 30-70 ethylene glycol-water; engine E.

Figure 30. - Effect of average coolant temperature on average cylinder-head temperature between exhaust valves. Coolant flow, 190 to 220 gallons per minute; engine coolant-outlet pressure, 10 to 35 pounds per square inch gage.
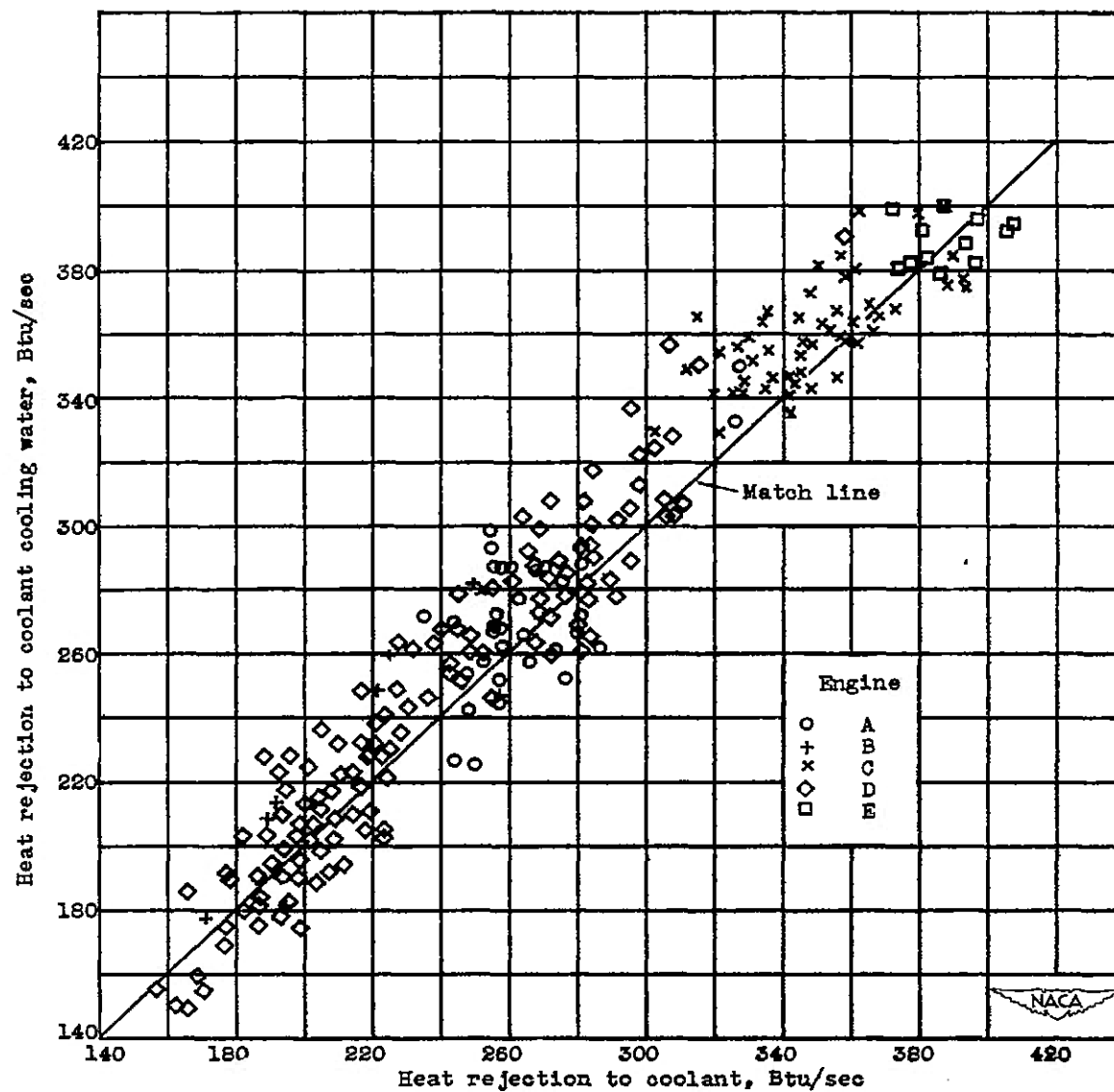
The following data table appears within the figure:

	Engine speed (rpm)	Manifold pressure (in. Hg abs.)	Charge flow (lb/ sec)	Fuel-air ratio	Carburetor-air temperature (°F)	Engine
o	2600	36	1.80	0.095	83	D
+	2600	37	1.84	.095	84	D
×	2600	44	2.21	.092	75	A
□	2600	44	2.21	.093	80	A
◇	3000	42	2.39	.095	50	B
△	3000	51	2.94	.095	50	B
▽	3000	50	3.40	.095	60	B

(a)   Coolant flow, 60 gallone per minute.

(b)   Coolant flow, 200 gallone per minute.

Figure 31. - Effeot of engine ooolant-outlet preeeure on cylinder-head
     temperature between exhauet valves when ueing 30-70 ethylene glycol-
     water as coolant at average coolant temperature of 226° F.   Engine
     power, 1800 brake horeepower; engine epeed, 3200 rpm; manifold
     preeeure, 70 inchee mercury aheolute; charge flow, 4.19 pounds per
     eecond; fuel-air ratio, 0.095; carburetor-air temperature, 0° F;
     engine E.

Figure 32. - Heat balance between coolant and coolant cooling water.

(a) Constant engine speed with variable manifold pressure.
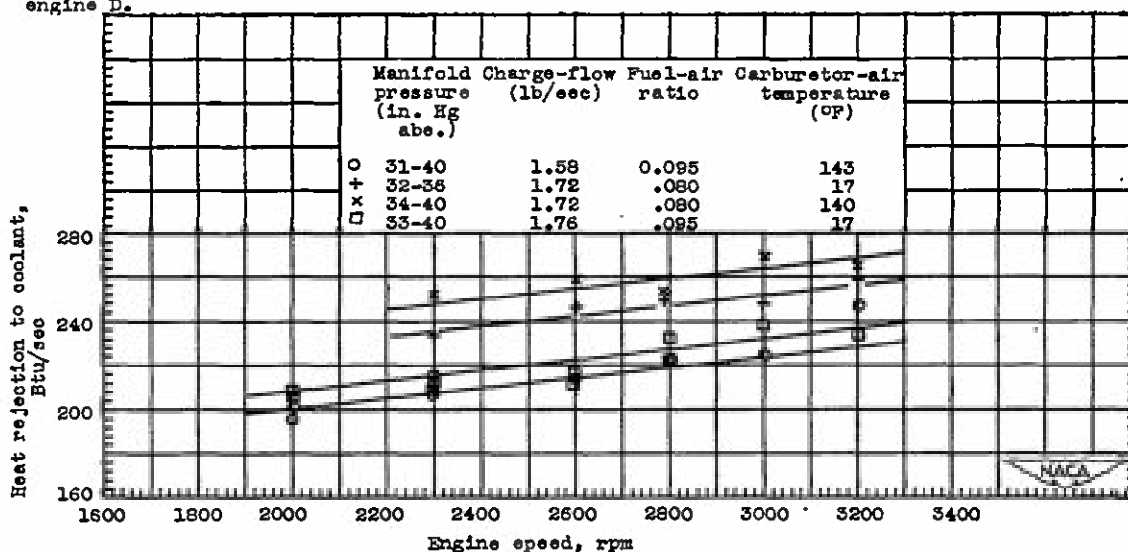
(b) Constant manifold pressure with variable engine speed.
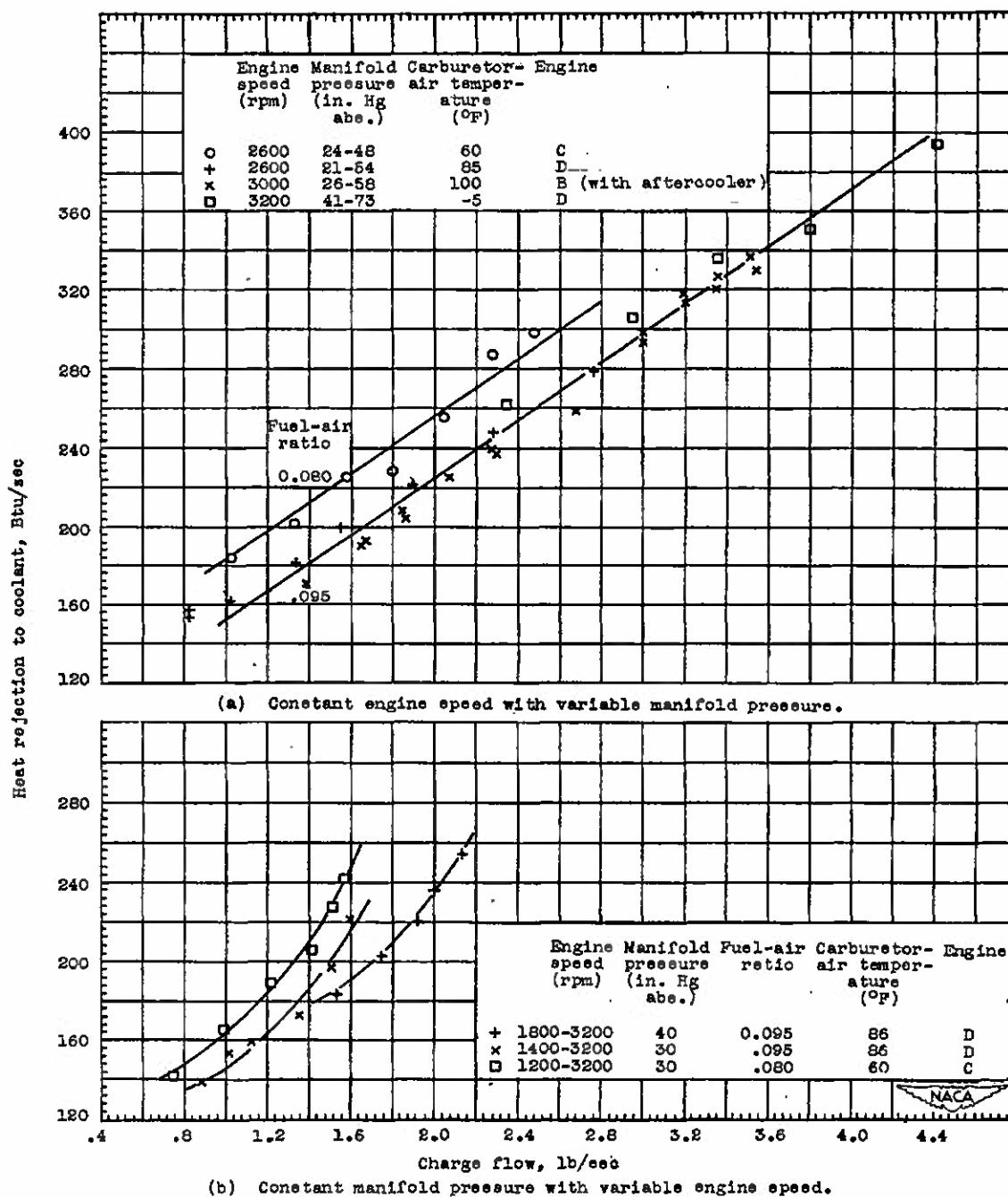
Figure 33. - Effect of charge flow on heat rejection to coolant. Coolant, 30-70 ethylene glycol-water; average coolant temperature, 245° F; coolant flow, 250 to 300 gallons per minute; engine coolant-outlet pressure, 13 to 36 pounds per square inch gage.
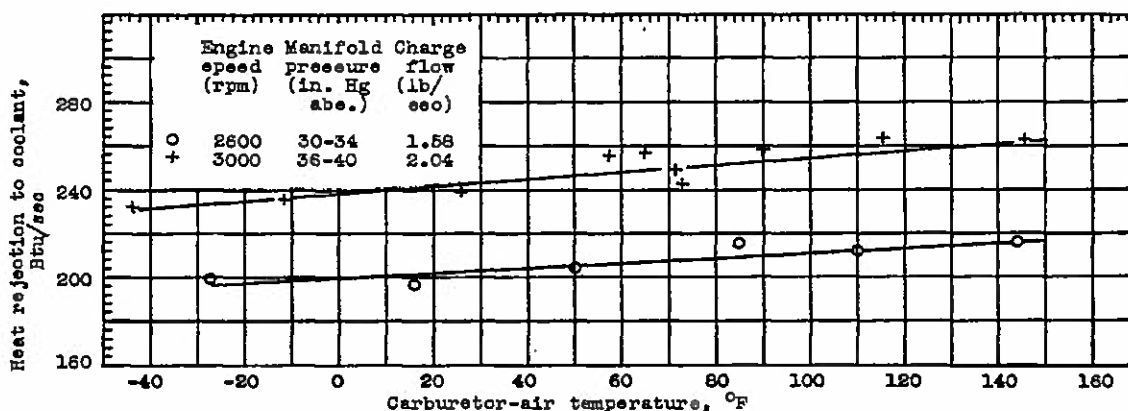
Figure 34. - Effect of carburetor-air temperature on heat rejection to coolant. Fuel-air ratio, 0.095; coolant, 30-70 ethylene glycol-water; average coolant temperature, 245° F; coolant flow, 300 gallons per minute; engine coolant-outlet pressure, 35 pounds per square inch gage; engine D.
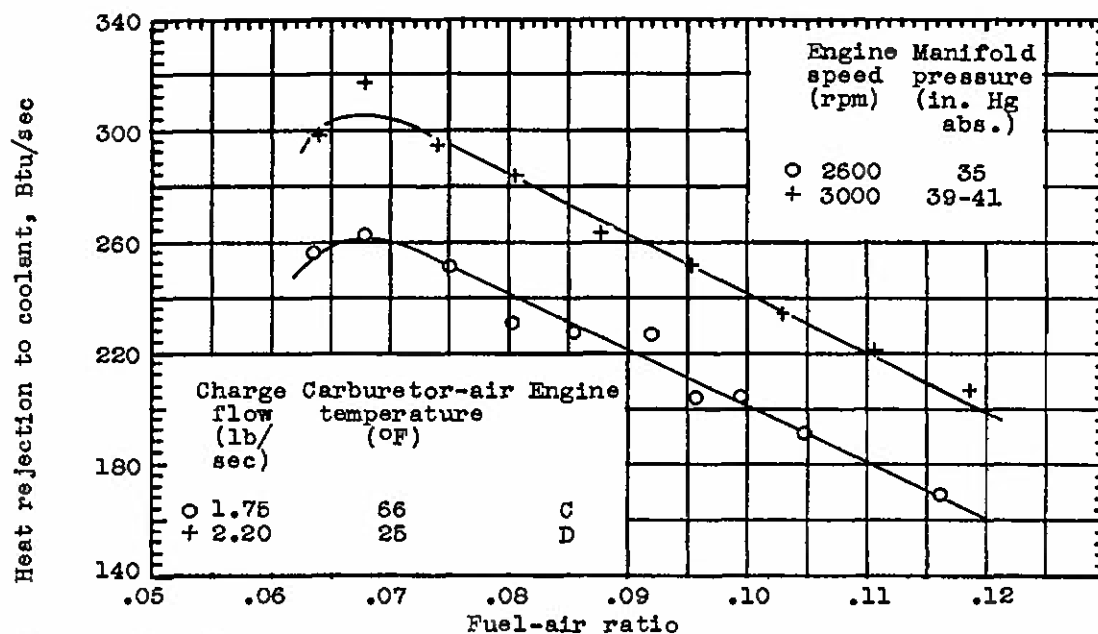

Figure 35. - Effect of engine speed on heat rejection to the coolant. Coolant, 30-70 ethylene glycol-water; average coolant temperature, 245° F; coolant flow, 300 gallons per minute; engine coolant-outlet pressure, 35 pounds per square inch gage; engine D.

Figure 36. - Effect of fuel-air ratio on heat rejection to coolant. Coolant, 30-70 ethylene glycol-water; average coolant temperature, 245° F; coolant flow, 220 to 270 gallone per minute; engine coolant-outlet preseure, 27 to 35 pounds per square inch gage.
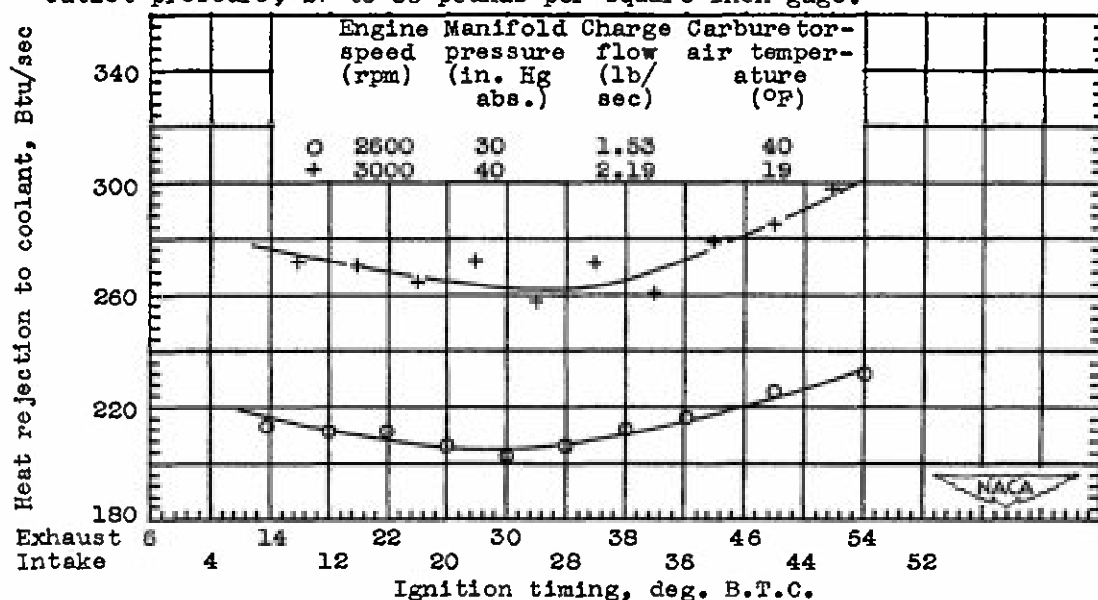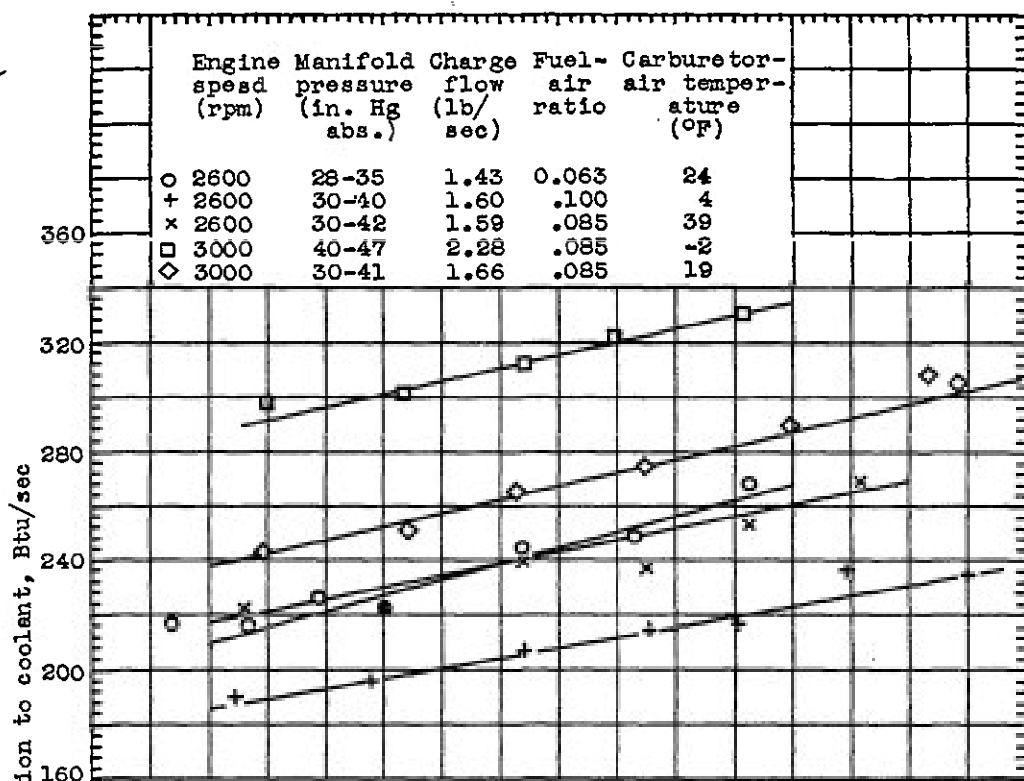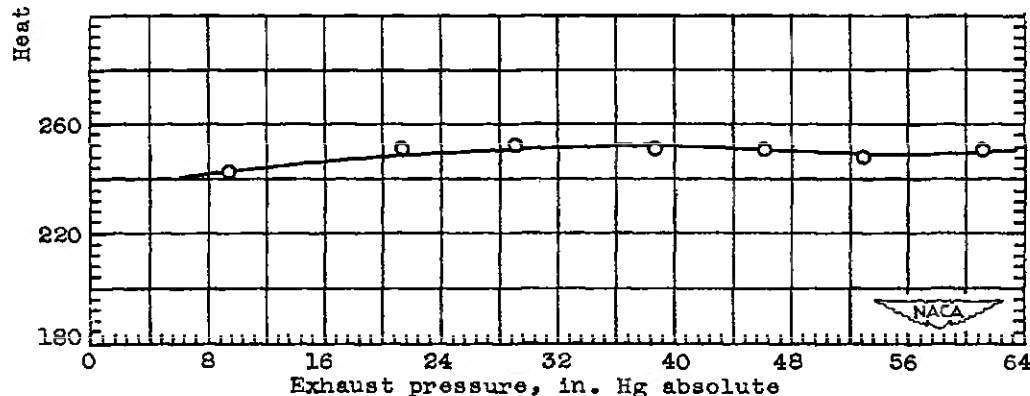

Figure 37. - Effect of ignition timing on heat rejection to coolant. Fuel-air ratio, 0.095; coolant, 30-70 ethylene glycol-water; average coolent temperature, 245° F; coolant flow, 300 gallons per minute; engine coolant-outlet pressure, 35 pounds per square inch gage; engine D.

Engine speed (rpm)	Manifold pressure (in. Hg abs.)	Charge flow (lb/sec)	Fuel-air ratio	Carburetor-air temperature (°F)
○ 2600	28-35	1.43	0.063	24
+ 2600	30-40	1.60	.100	4
× 2600	30-42	1.59	.085	39
□ 3000	40-47	2.28	.085	-2
◇ 3000	30-41	1.66	.085	19

(a)   Constant charge flow with variable manifold pressure.

(b)   Constant manifold pressure with variable charge flow:
Engine speed, 2600 rpm; manifold pressure, 35 inches
mercury absolute; charge flow, 1.36 to 1.92 pounds
per second; fuel-air ratio, 0.085; carburetor-air
temperature, 32° F.

Figure 38. - Effect of exhaust pressure on heat rejection to coolant.
Coolant, 30-70 ethylene glycol-water; average coolant temperature,
245° F; coolant flow, 250 gallons per minute; engine coolant-outlet
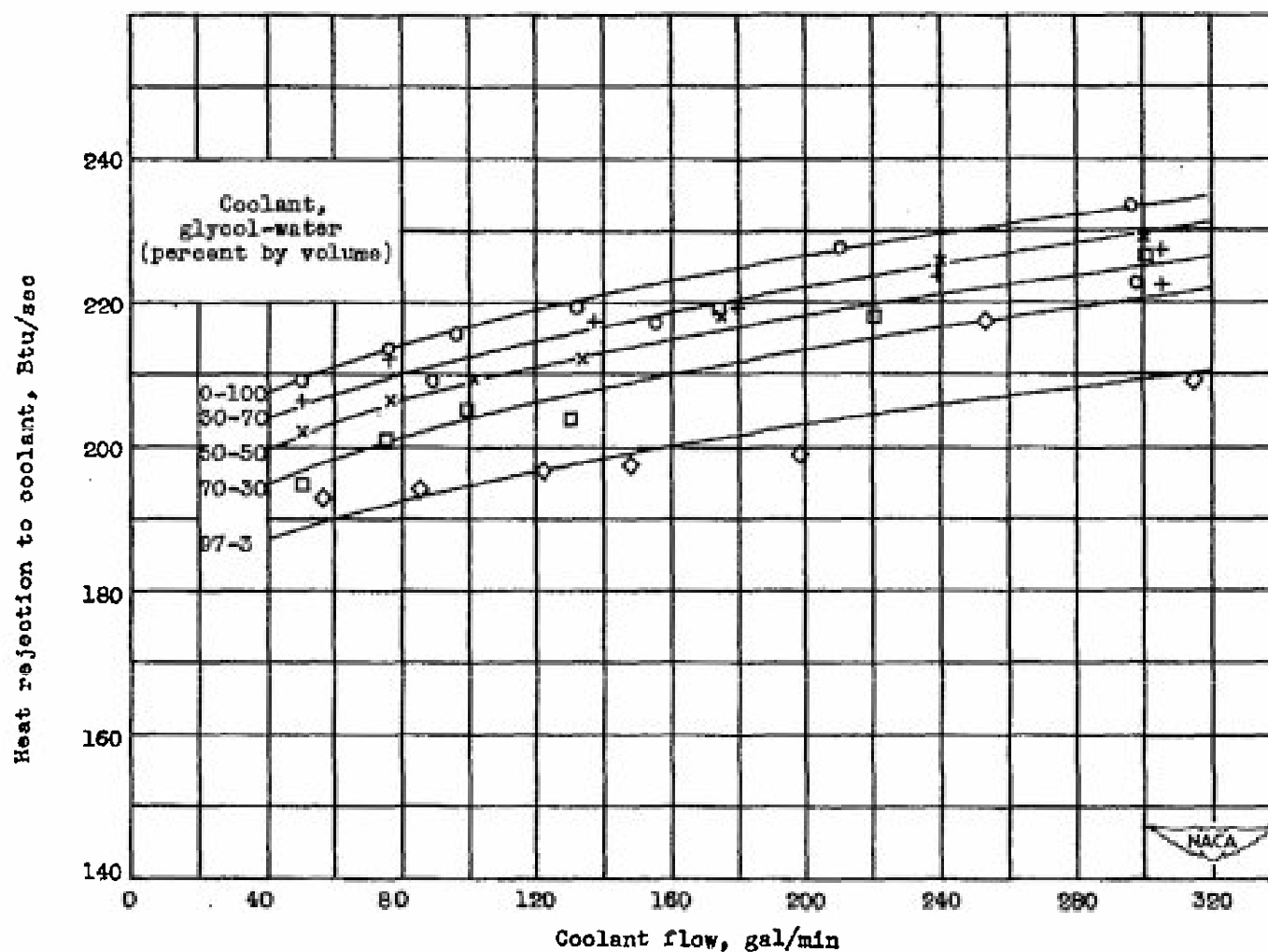pressure, 35 pounds per square inch gage; engine D.

Figure 39. – Effect of coolant flow on heat rejection to coolant. Engine power, 780 brake
horsepower; engine speed, 2600 rpm; manifold pressure, 36 inches mercury absolute; charge
flow, 1.78 pounds per second; fuel-air ratio, 0.095; carburetor-air temperature, 83° F;
average coolant temperature, 245° F; engine coolant-outlet pressure, 35 pounds per square
inch gage; engine D.

(a)   Coolant-outlet temperature, 250° F.
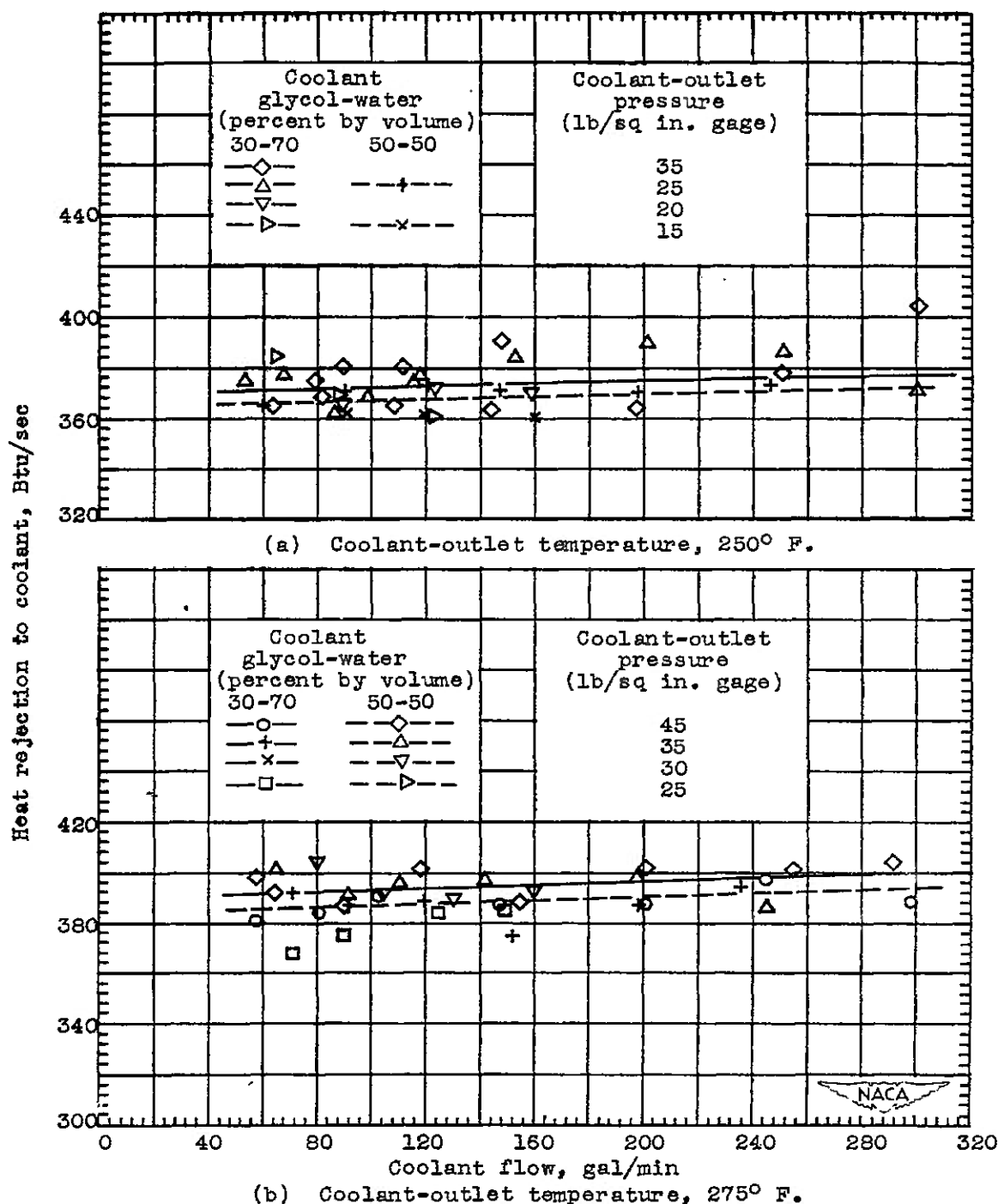
(b)   Coolant-outlet temperature, 275° F.

Figure 40. - Effect of coolant flow on coolant heat rejection. Engine power, 1800 brake horsepower; engine speed, 3200 rpm; manifold pressure, 70 inches mercury absolute; charge flow, 4.19 pounds per second; fuel-air ratio, 0.095; carburetor-air temperature, 0° F; engine C.
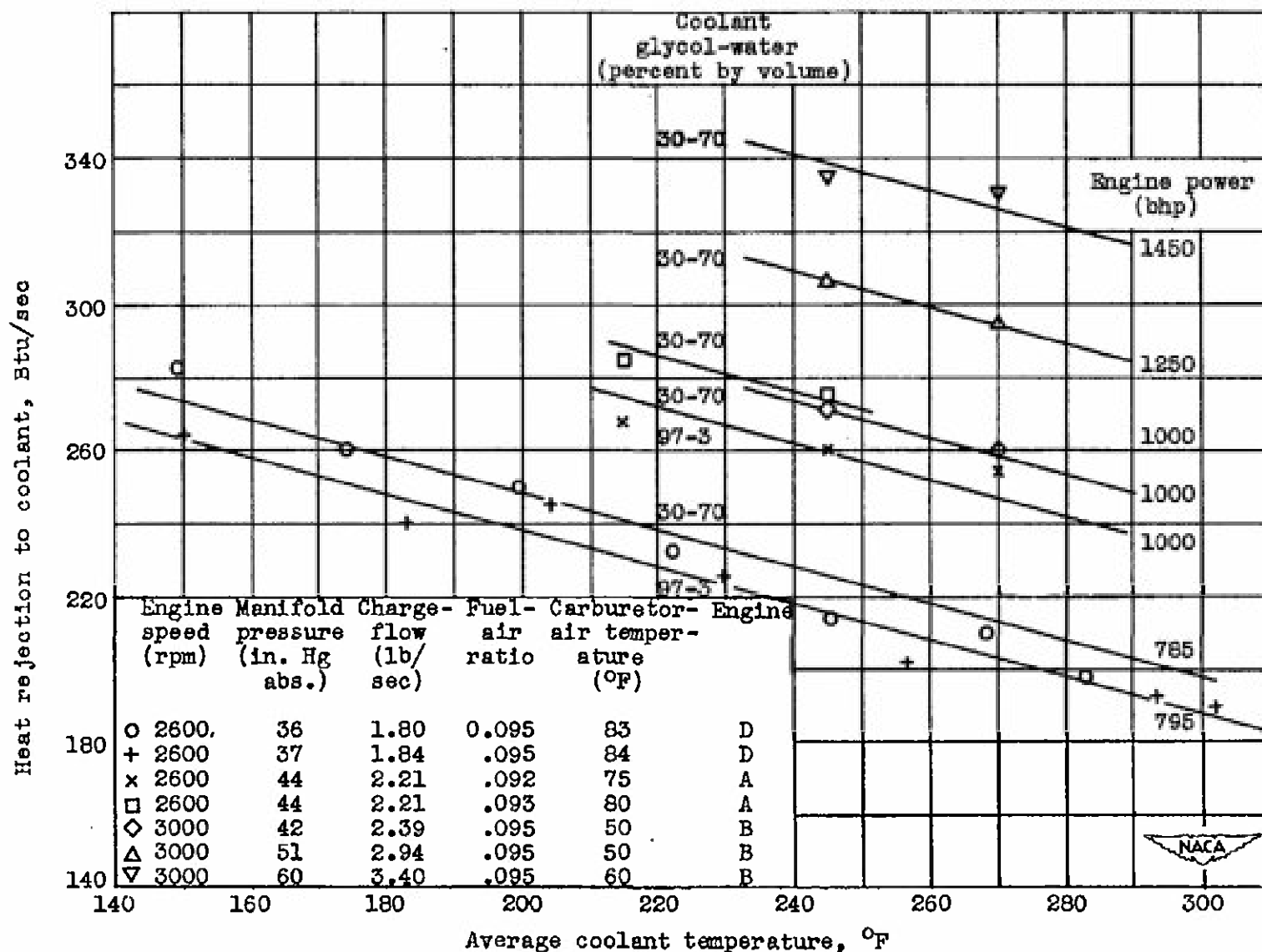
Figure 41. - Effect of average coolant temperature on heat rejection to coolant. Coolant flow, 190 to 220 gallons per minute; engine coolant-outlet pressure, 10 to 35 pounds per square inch gage.